Design and Evaluation of Potential Cooling Cycles for the Reduction of Water Use in Thermoelectric Power Plants

Thesis

By

Rhys Davis

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Thesis Committee:

Dr. Shaurya Prakash, Adviser

Dr. Bhavik Bakshi

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119 Hitchcock Hall

College of Engineering

The Ohio State University

Columbus, Ohio 43210

Abstract

In the U.S., water for thermoelectric power plant cooling constitutes 40% of all freshwater withdrawal and 4% of all freshwater consumption. There are two main kinds of power plant cooling systems, once-through and recirculating, but environmental issues arise from both types, including excessive water consumption and ecological damage from high temperature return water. The purpose of this study is to identify effective while environmentally safe ways to reduce water use at power plants and determine the water savings and potential cost savings resulting from these methods. The strategy to reduce cooling water use that was employed in this study was to lower the temperature of the cooling water at either the inlet or outlet of the working steam condenser. This would allow for a higher temperature differential for the cooling water over the condenser, which, for the same cooling load, would mean a lower water flow rate is required. To achieve this, multiple refrigeration cycles were designed within constraints based on environmental concerns and maximum energy inputs. These cycles were then analyzed for the two types of cooling systems and four different regions in the United States. The most beneficial refrigeration cycle was chosen for each region, and the available cooling from the refrigeration cycles was used to determine water savings for each situation. The most effective refrigeration cycle offers savings of around 3% of the initial flow rate, which can correspond to around 4 billion gallons of water withdrawal savings per year for once-through systems and around 80 million gallons of consumption savings for recirculating systems. This could correspond to millions of dollars of savings per year at a single plant, and this will only increase as demand for water increases and supply continues to decrease. The findings from this research will identify the types of cooling systems that will be beneficial for different regions and power plant types, and can be used to further develop these systems at a larger scale, along with identifying other water saving strategies that merit more research.

Acknowledgements

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1. Introduction

In the United States, nearly 90% of electricity generation is produced by thermoelectric power plants [1]. Thermoelectric power plants function by using a working heated fluid, almost always steam, to spin a turbine, which drives a generator. The heat to drive this process can come from any source of fuel, but it is almost always coal, natural gas, or nuclear energy. The working fluid must be cooled before being cycled through the turbine again, and virtually all power plants use water to absorb the waste heat from the working fluid that was not able to be used to spin the turbine. There are two main forms of power plant cooling systems that use water as the refrigerant: once-through cooling, which includes pond cooling, and recirculating cooling.

Once-through cooling systems obtain water to cool the working fluid by withdrawing water from a source, such as a lake, river, ocean, or man-made pond, as shown in Figure 1. After the cooling water absorbs the unused heat from the working fluid in the power plant, it is then returned to the source at an elevated temperature. There are regulations that limit what temperature the water can be returned to the source at, so this type of cooling system requires a large withdrawal of water to effectively remove the heat from the system without raising the temperature of the cooling water than more than about 15°C [2]. These types of systems account for about 43% of thermoelectric electricity generation [3].

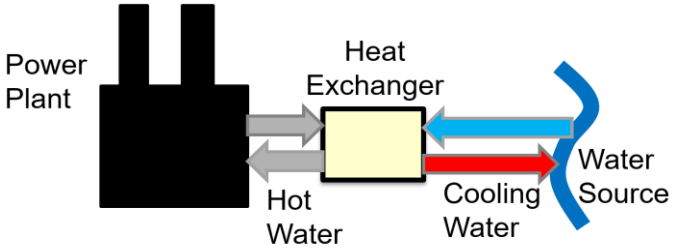

Figure 1: Once-through cooling water removes unused waste heat from working fluid and is returned to water source with heat

Recirculating cooling systems were developed so that power plants could be effectively cooled without needing to withdraw a significant amount of water from a local source. Instead, water is recirculated, using a cooling tower to remove heat from the cooling water so it can be sent back to the working fluid condenser. This can be observed in

Figure 2. When the cooling water is in the cooling tower, a percentage of the water, usually around 2% [4], is lost to evaporation. Although the percentage lost is small, for large power plants in water stressed areas, this is still a significant amount of water, as much as 7.5 million gallons a day, that must be made up by drawing from a local water source. These types of systems account for about 54% of thermoelectric electricity generation [3].

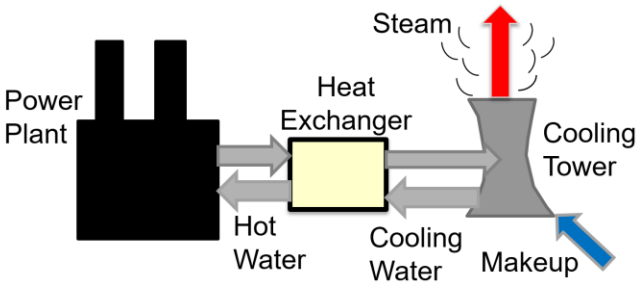

Figure 2: Recirculating cooling water removes unused waste heat from working fluid and rejects heat to ambient air before being recycled

Water use for thermoelectric power plants is the largest cause for water withdrawal in the U.S. On a given day, 138 billion gallons of water are withdrawn, and 4.3 billion gallons are consumed, for use as power plant cooling water. This accounts for 40% of U.S. freshwater withdrawal and 4% of consumption, respectively [5]. The water that is withdrawn and returned is typically discharged at around 8 to 12°C higher than the intake temperature, although some systems have discharged at around 15°C higher. This can have disastrous ecological effects, such as killing many kinds of fish species, reducing oxygen solubility in the water source, and encouraging unchecked algal growth, also known as algal blooms [6]. States generally limit water discharge temperature to 32°C or lower, but in recent years, this limit has been often exceeded [2]. With the cooling water return temperature limited by regulations, increased inlet temperatures can have negative effects for power plants. First, warmer cooling water can decrease the efficiency of a plant. Plant efficiency is mostly related to the working fluid pressure drop over the turbine, so a lower pressure in the condenser means a higher thermal efficiency. The lower the temperature of the cooling water entering the condenser, the lower the temperature

and thus saturation pressure of the working fluid can reach [7]. Second, if the cooling water temperature differential over the condenser is decreased due to a higher inlet temperature and outlet temperature bounded by regulations, more water withdrawal will be required. In some cases, plants will not be able to provide enough cooling or will be forced to shut down due to either too high of inlet or outlet cooling water temperatures, like Browns Ferry nuclear plant in Tennessee, which, in 2010, had to drastically cut its energy output in middle of a heatwave in peak energy demand season because the water source it was drawing from became too hot [8].

Not all plants use cooling water in the same way, as illustrated above. The most helpful way to measure water use at power plants is how much is used in relation to how much energy is produced: once-through cooling generally withdraws around 35,000 gallons of water per megawatt-hour of electricity produced (gal/MWh) and consumes around 200 gal/MWh; recirculating cooling generally withdraws around 1,000 gal/MWh and consumes around 600 gal/MWh. These values also change based on the fuel source and plant type. For a more detailed breakdown of water use in different power plant types, see Table 1 below [9]. The "generic" category for cycle technology in the table refers to the industry standard cycle, which is a combined cycle, or a Brayton cycle combined with a Rankine cycle.

Table 1: Water consumption and withdrawal for different fuel types, cooling types, and cycle technology type

Fuel Type	Cooling	Technology	Median Consumption (gal/MWh)	Median Withdrawal (gal/MWh)
Nuclear	Tower	Generic	672	1,101
	Once-through	Generic	269	44,350
	Pond	Generic	610	7,050
Natural Gas	Tower	Combined Cycle	198	253
		Steam	826	1,203
		Combined Cycle with CCS	378	496
	Once-through	Combined Cycle	100	11,380
		Steam	240	35,000
	Pond	Combined Cycle	240	5,950
	Dry	Combined Cycle	2	2
Coal	Tower	Generic	687	1,005
		Subcritical	471	531
		Supercritical	493	609
		IGCC	372	390
		Subcritical with CCS	942	1,277
		Supercritical with CCS	846	1,123
		IGCC with CCS	540	586
	Once-through	Generic	250	36,350
		Subcritical	113	27,088
		Supercritical	103	22,590
	Pond	Generic	545	12,225
		Subcritical	779	17,914
		Supercritical	42	15,046

Once-through cooling systems have obvious negative effects on local water sources, as mentioned above. Recirculating cooling systems can also have a negative effect on the local ecosystem, however, depending on the region. Recirculating power plants tend to be found in the regions with a lack of water; 59% of power plants in the most water stressed regions are recirculating, compared to 49% of power plants in the least water stressed regions [10]. This means that recirculating systems, which consume more water than once-through systems, can remove valuable water from very strained sources and relocate it to a more distant watershed through evaporation.

There has been a significant amount of research done and technology developed to combat the problem of water use at thermoelectric power plants. One of the more prominent strategies is to use a dry cooling system; this type of cooling currently accounts for 2% of all power plant cooling. This system implements a step in-between the cooling water and dissipating the heat to the environment by using a closed loop system and a dry cooling tower to transfer heat to the ambient air. This is known as indirect dry cooling; direct dry cooling removes the cooling water altogether and uses air to cool the working fluid. The downsides of dry cooling that make implementing them problematic and currently unrealistic are the capital and energy costs. Dry cooling requires a completely different, closed system compared to once-through or recirculating systems, so to implement a dry cooling system, an entirely new cooling tower must be built, which is a prohibitive capital cost unless a new power plant is being built or the existing tower needs to be replaced. Dry cooling uses air instead of water to remove waste heat from the power plant, and air has a specific heat that is less than 25% of water's specific heat. A lower specific heat means that for the same mass and temperatures, less heat will be able to be removed from the system. Also, ambient air generally has a higher maximum temperature than most water sources, especially in dry climates, which means that the temperature differential between the ambient air and the working fluid is less, requiring more air to be drawn through and, thus, more energy. For example, the average air temperature in Texas is about 5°C higher than the average surface water temperature, which means that there will be many days where dry cooling is much more energy intensive than wet cooling [11, 12]. Dry cooling also reduces the efficiency of the cooling process because less heat can be expunged from the system, so the amount of electricity that can be produced from a unit of fuel is reduced as well, increasing energy costs [10].

There are many other ideas that attempt to improve on some of the drawbacks of once-through, recirculating, and even dry cooling that have been researched and tested, although none have been implemented on a large scale. These ideas stem from many different spaces in the water-energy nexus, like improved materials, processes, and technologies. One area that is the subject of much research is more efficient heat transfer in heat exchangers, which would create a more efficient system that would reduce the amount of water necessary to capture the heat from the working fluid. This can be achieved through the introduction of novel materials into the heat exchangers that have extremely high thermal diffusivities. Another promising area of research is using nanofluids that have higher heat-carrying capacities than water to remove heat from the working fluid; unfortunately, this technology has not passed the lab stage [5].

Ultimately, the most effective solution to water and ecological stress caused by power plant water use will be a mix of different technologies and systems that can be implemented at different sizes and types of plants. This will require systems that can be implemented with relatively low capital costs compared to the cost of a new cooling tower, and must use technologies that are ready to be implemented at scale in the immediate future. It will be important to identify what systems will be best suited for different regions, and how they will affect efficiency, energy costs, and the surrounding environment.

2. Overview of Thesis

As discussed in the previous section, there are many different areas that have been identified that could potentially help lower power plant water use. This study will focus solely on reducing the cooling water temperature, which will reduce the quantity of cooling water needed for the same quantity of heat being removed from the plant. There are two strategies for reducing

the temperature of the cooling water: using a refrigeration cycle at either the inlet or the outlet of the hot water condenser or using water from a cooler source.

The purpose of this research is to identify the best strategy for lowering the temperature of cooling water for different types of plants in different regions. It is important for the recommended strategy to have a low environmental and ecological impact, while not drastically increasing the cost of electricity for the plant. In the following sections, the potential refrigeration cycles will be chosen and optimized, water temperature from different sources and regions will be quantified, and a matrix of water savings and costs will be assembled for different regions and plant cooling types.

3. Methodology

3.1. Assumptions

Before determining the refrigeration cycles and water sources to be analyzed, assumptions about the power plant system and surroundings had to be made. The two plant cooling types that will be analyzed are recirculating and once-through, because they make up about 97% of all cooling types and are the two main types that use water [3]. A standard power plant size of 500 MW will be used for analysis. This is a standard size for a large gas or coal power plant, per the Electric Power Research Institute (EPRI) [13]. Power plant and water source regions will be divided into four areas: Northeast (New England, New York, Pennsylvania, New Jersey, Delaware, Maryland, Virginia), Midwest (All states bordering the Ohio River, Missouri, Iowa, Minnesota, Wisconsin, Michigan), Southeast (All states enclosed by, and including, Louisiana, Arkansas, Tennessee, and North Carolina), and West (All states west of the states bordering the Mississippi River). Under the geographic classifications made here, the Northeast makes up 15% of the United States' electricity generation, the Midwest makes up 23%, the Southeast makes up 27%, and the West makes up 36%, although that number is skewed by

Texas, which alone makes up 12% of the country's electricity generation [14]. The standard temperature difference across the working fluid condenser, without any additional refrigeration of the cooling water, is assumed to be 10°C, based on EPRI studies and an analysis by Madden *et al.* [13] [2]. For recirculating plants, the required temperature difference between the cooling water leaving the cooling tower and the ambient air entering the tower, also called approach, will be 5°C, which is a relatively standard value [15].

3.2. Constraints

Next, constraints were developed based on research and literature on power plant cooling and tangential subjects. In any strategy evaluated, the cooling water must be returned to either the cooling tower or original source water, depending on the type of cooling, at the temperature it was returned at prior to the refrigeration cycle being introduced. Energy used for work in the refrigeration cycles cannot exceed 2% of the total energy output of the plant; in this case, that means 10 MW. This is a rough metric based on the operation and maintenance costs for a typical coal or gas plant, which, annually, are about 2% of the overnight capital cost of the plant [16]. Energy used for heat in the refrigeration cycles cannot exceed 6% of the total energy output of the plant; in this case, that means 30 MW. This metric is based on the amount of useful residual heat generation for a given plant size; a plant generates about 1.5 times as much residual heat as power for electricity, and about 4% of this heat is at a high enough temperature to be used by a generator [17]. All refrigerants used should have a global warming potential (GWP) of 10 or lower and an ozone depletion (ODP) of 0, so as not to harm the environment in case of a leak. With these assumptions and constraints in mind, a literature study was performed on different industrial cooling cycles and refrigerants.

3.3. Refrigeration Cycles and Refrigerants

The simplest and most common refrigeration cycle is a single stage vapor compression refrigeration cycle. There are four components: an evaporator, where the refrigeration takes place, a compressor, a condenser, and an expansion valve. The refrigerant spends most of the cycle inside of the saturation curve, meaning it is both a liquid and a vapor, but will become a superheated vapor. The only energy input for this cycle is the work needed for the compressor [18].

For colder refrigeration constraints, a cascading vapor compression refrigeration cycle is a more effective option than a single stage cycle. A two-stage cascading refrigeration cycle uses two compression cycles; one cycle, the low temperature cycle, extracts heat from the fluid or area being cooled and the second cycle, the high temperature cycle, extracts heat from the low temperature cycle. This cycle can achieve lower temperatures than the single stage cycle because it splits the cooling load up into two cycles, which eases the strain put on the compressor by the larger pressure gradient created by the larger temperature differential [19].

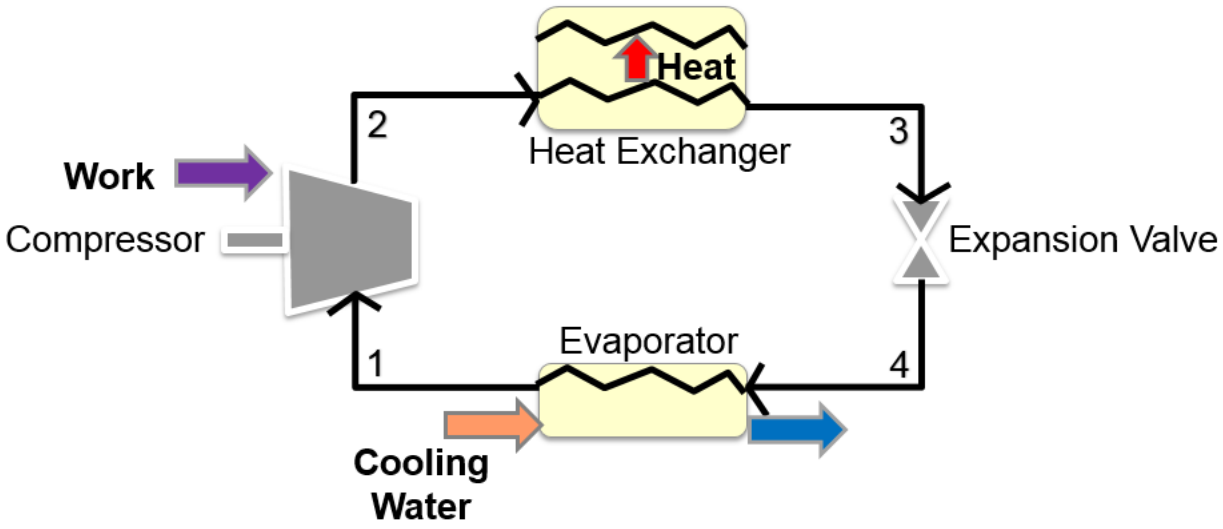

Figure 3: Typical compression refrigeration cycle that uses compressor work as an energy input

Another refrigeration cycle that is commonly used is an absorption refrigeration cycle. Three of the four components of the compression cycle are used in the absorption cycle, namely the condenser, expansion valve, and evaporator. However, in this cycle, the compressor is replaced by a more complex system where the refrigerant is combined with a weak solution of an absorbing fluid, pumped up to a generator where heat is added, and the refrigerant dissociates from the absorbent before being sent into the condenser. This cycle requires virtually no energy for input work, only the work for the pump. Instead, almost all the energy required for this cycle is in the form of heat for the generator. This cycle can be cascaded with a compression cycle to reach a lower temperature for refrigeration [18].

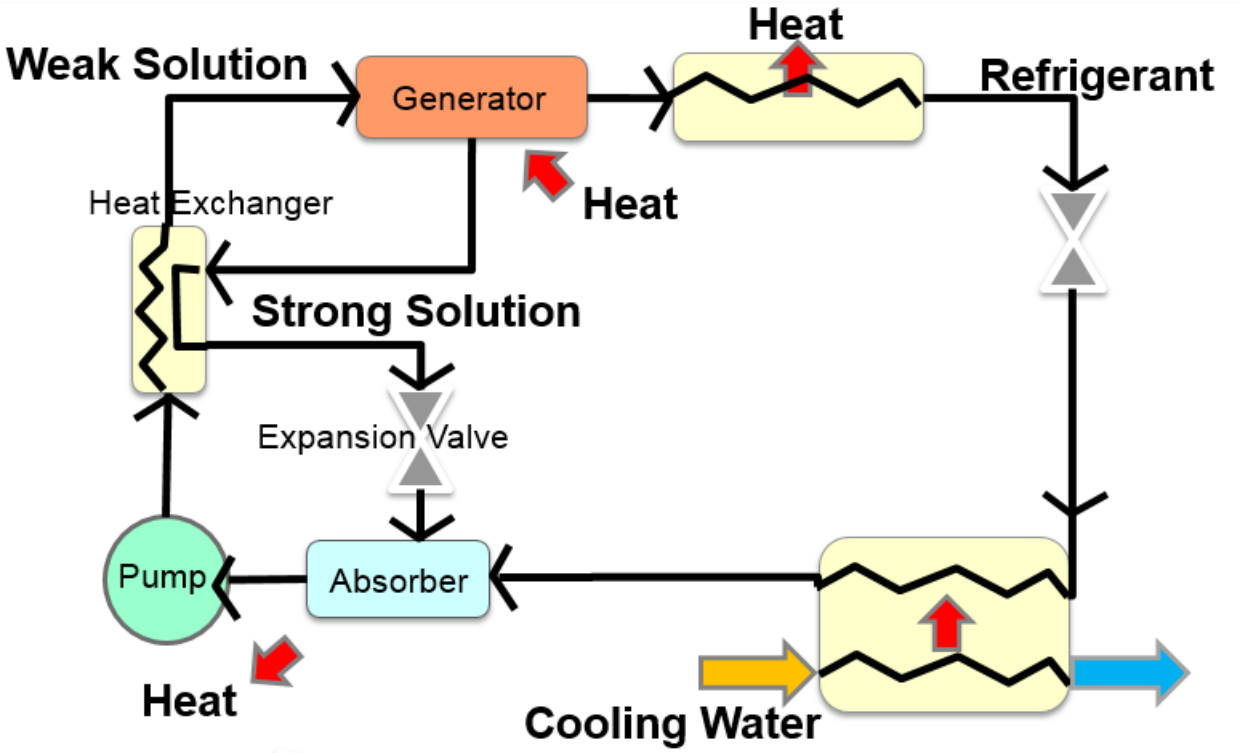

Figure 4: Typical absorption refrigeration cycle that uses generator heat as energy input

Other refrigeration cycles that can be used but are uncommon or not ideal for larger, cooler applications are essentially cycles used for work or heat output, but in reverse, like the reverse Carnot and reverse Stirling cycle. These do not have much application for this research.

The list of refrigerants that could be used for these cycles based on the constraints of GWP and ODP is relatively small. Nearly all refrigerants currently or recently in use are one of three varieties: chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs), and hydrofluorocarbons (HFCs). CFCs have the highest GWP and ODP of any of the groups and have mostly been banned; HCFCs have slightly better GWPs but still have a nonzero ODP; HFCs, introduced as an alternative to CFCs and HCFCs, have an ODP of zero, but still have very high GWPs. This leaves natural or more commonly produced refrigerants with a zero ODP, namely carbon dioxide (CO_2, or R744), which has a GWP of 1; ammonia (NH_3, or R717), which has a GWP of 0; and petroleum gases like butane (C_4H_{10}, or R600) and propane (C_3H_8, or R290), which have GWPs of 3. Water can also be used as a refrigerant in an absorption refrigeration cycle with the appropriate absorber fluid [20].

3.4. Modeling and Data Gathering

After determining the appropriate cycles and refrigerants to analyze based on the energy requirements of the cycles and the thermodynamic properties of the refrigerants, the next step will be to optimize the cycles so that they can be appropriately compared. To do this, the cycles will be setup in Engineering Equation Solver (EES) and, using information from prior research, some parameters will be varied and a curve will be produced showing how these parameters effect COP and energy input.

Next, the water sources and respective temperatures must be determined. To do this, the United States Geological Survey's (USGS) data for surface water quality was used to estimate an average of lake, water, and reservoir temperatures for the states that have the largest electricity generation in their respective region [12]. This meant using Pennsylvania's data for the Northeast, Illinois's data for the Midwest, Florida's data for the Southeast, and Texas's data for

the West [14]. To estimate the temperature in aquifers and shallow ground water for each region, a map of the average temperatures nationwide was used [21].

The cycles must be analyzed for all combinations of the following options: recirculating or once-through cooling, surface or aquifer cooling water, and the different regions. To analyze cycles in a recirculating plant, the properties of a cooling tower must be established. The ambient temperature and relative humidity of the air will be determined for each region by taking an average from a selection of cities in the states mentioned in the previous paragraph, Pennsylvania, Illinois, Florida, and Texas, using Global Climate Station Summaries from the National Climatic Data Center (NCDC) [11]. From this and the assumptions that the water temperature must decrease 10°C across the cooling tower and approach must be 5°C, the required makeup water can be determined through a formula from Perry's Chemical Engineers Handbook based on evaporation, drift, and blowdown loss [22]. The outlet temperature will be a weighted average, based on flow, of the temperature of the makeup water and the temperature of the cooling water coming out of the cooling tower. The water savings associated with this cycle will be a decrease in consumption, and will be determined by the differences in makeup water required for the system. For once-through plants, the water must be returned at 10°C higher than the intake water temperature. The savings associated with this type of cycle will be a decrease in withdrawal; it is true that it may be more environmentally impactful in some regions to reduce the temperature of the return water instead of reducing the withdrawal, but for the sake of comparison, savings will be kept in terms of water, not temperature reduction.

To analyze the cycles for the different regions and source water options, the only thing that will change is source water temperature, which will be determined using the methods described previously.

All the cycles will be restricted to not exceeding 2% of the total plant generation for input work, 10 MW in this case, or 6% of the total plant generation for input heat, 30 MW in this case. From this, the maximum water savings, either consumptive or withdrawal, can be determined. After determining this for all the combinations of conditions, the avoided cost of water withdrawal will be determined by a formula developed by Ulrich *et al*. to calculate power plant water costs based on flow, consumer price index, and fuel prices [23]. This means that water savings may be appropriately captured economically for once-through systems, but not for recirculating systems, because the cost of consumed water cannot be differentiated from withdrawn water with this equation. The equation is not a perfect estimation, either, and will vary widely based on region, fuel type, and other characteristics of the plant, but as a rough estimate to compare cost savings and get an idea of the magnitude of savings, it will suffice. Finally, the capital and energy costs for the refrigeration cycles will be determined to understand the economic viability of these cooling strategies.

After all the thermodynamic and economic calculations have been performed, a refrigeration strategy will be recommended for each region and type of cooling, and the economic viability of that cycle will be discussed.

4. Results

4.1. Refrigeration Cycles

After considering the constraints and assumptions listed in the Methodology section, four refrigeration cycles were chosen to analyze. For these cycles to realistically cool large quantities of water with existing refrigeration technology, it was decided that all cycles should be cascading, with a low temperature and high temperature cycle. This allows the cycles to split the cooling load, reducing the amount of work or heat input needed for either cycle. Of the refrigerants available based on the constraints, the best low temperature refrigerant is carbon dioxide (CO_2). The saturation curve for CO_2 is shown in Figure 5. It has a uniquely low critical temperature of 31.1°C, which means it becomes a supercritical vapor above this point. As a comparison, R-134a, a common refrigerant, has a critical temperature of 101.1°C. This is not normally a beneficial trait for a refrigerant to have, because it is harder to use in a subcritical compression cycle, which is more ideal than a transcritical or supercritical cycle, because ambient air may not be cool enough to condense it below its saturation point. However, used on the low temperature side of a cascading refrigeration cycle, it can be evaporated at a lower temperature by the high temperature cycle

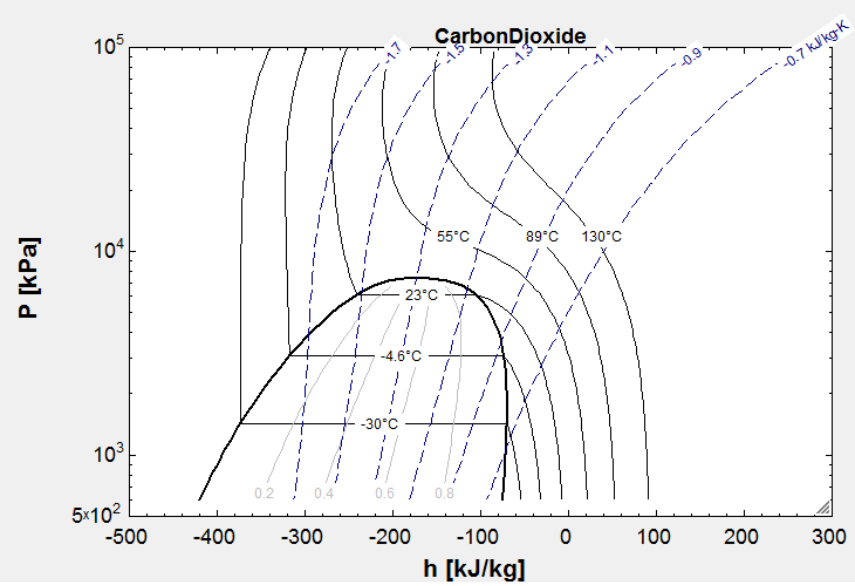

Figure 5: CO_2 saturation curve on pressure vs. enthalpy chart

[24]. Thus, for all cycles, there is a low temperature subcritical CO_2 compression refrigeration cycle.

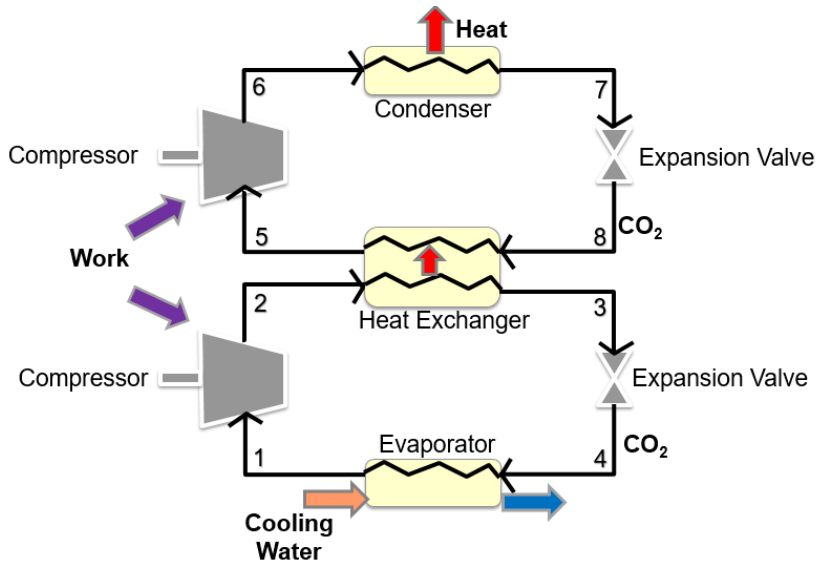

Figure 6: CO_2 – CO_2 cascading compression refrigeration cycle diagram

The first high temperature cycle analyzed was simply another CO_2 compression cycle, but this one was a transcritical cycle. This is less ideal than a subcritical cycle, but because of its availability, low cost, and relative harmlessness as a gas, it is a good refrigerant to use for this application. The diagram for this cycle is shown in Figure 6.

The next high temperature cycle analyzed was an ammonia (NH_3) compression cycle. The saturation curve is shown in Figure 7. The critical temperature of NH_3 is 132.4°C, so it is much better suited as a high temperature refrigerant for the compression cycle than CO_2, but it is highly toxic [25]. The diagram is shown in Figure 8.

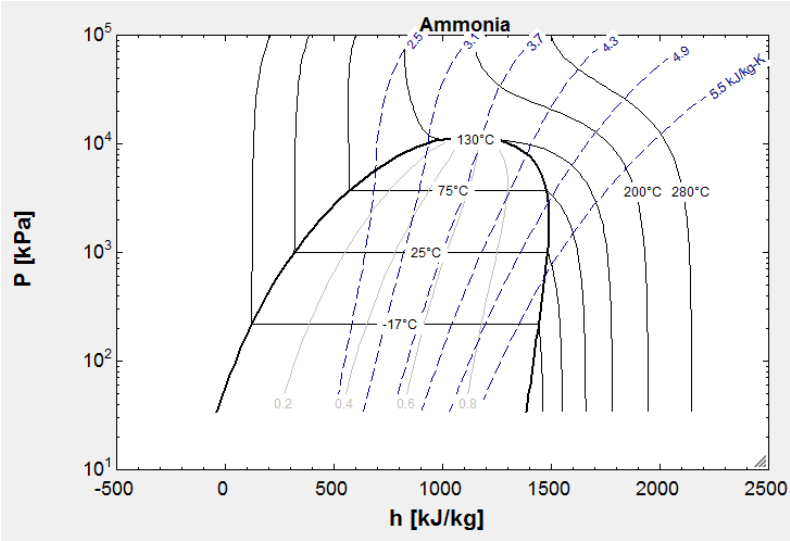

Figure 7: NH_3 saturation curve on pressure vs. enthalpy chart

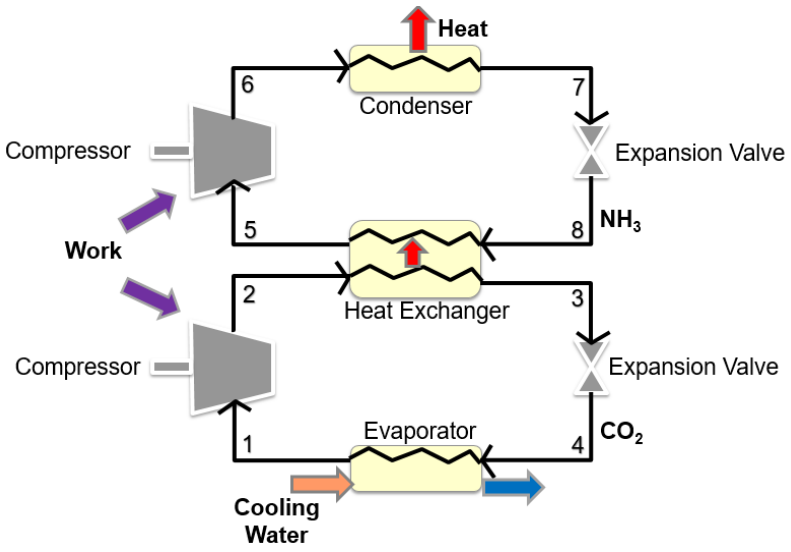

Figure 8: CO_2 – NH_3 cascading compression refrigeration cycle diagram

The next high temperature cycle analyzed was an absorption refrigeration cycle, a lithium bromide and water (LiBr-H_2O) absorption cycle. As mentioned in the Methodology section, absorption cycles are beneficial for refrigeration in facilities that produce a large amount of waste heat, such as thermoelectric power plants. The benefit of using LiBr and water is that the vapor of the water can almost completely dissociate from the LiBr in the generator, which means that there is no rectifier necessary to rid the absorbent, LiBr, from the refrigerant, water, after heat is added in the generator [26]. The diagram of the cycle is shown in Figure 9.

The final cycle that was analyzed was a water and ammonia (H_2O-NH_3) absorption refrigeration cycle, where water is the absorbent and NH_3 is the refrigerant. This cycle has the same components as the LiBr-H_2O cycle, except that it needs a rectifier after the generator to remove the water vapor from the ammonia. The benefit of using the

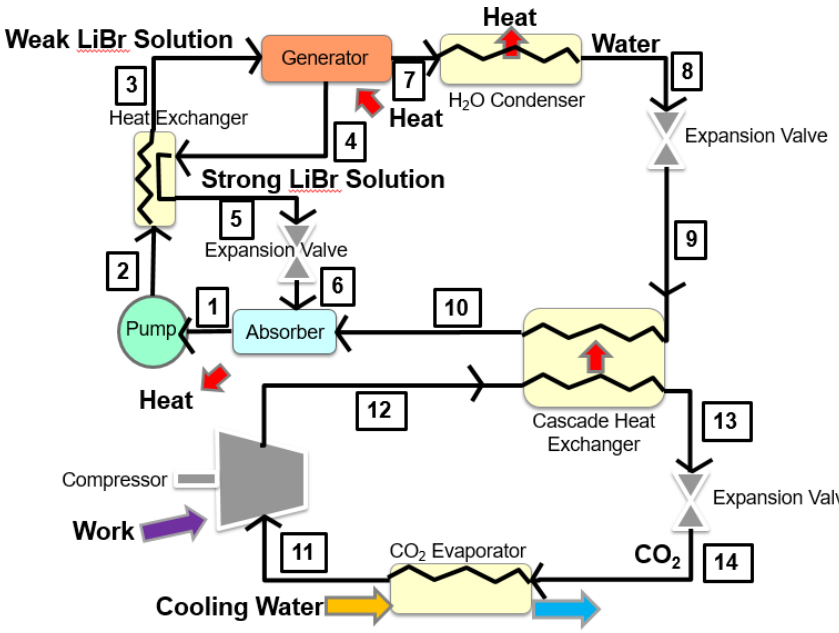

Figure 9: CO_2 - LiBr-H_2O cascading compression and absorption refrigeration cycle diagram

H_2O-NH_3 cycle is that it can operate at lower temperatures than the LiBr-H_2O cycle because water is limited as a refrigerant to 0°C, and LiBr will recrystallize at a certain temperature, restricting flow [27]. The diagram of the cycle is shown in Figure 10.

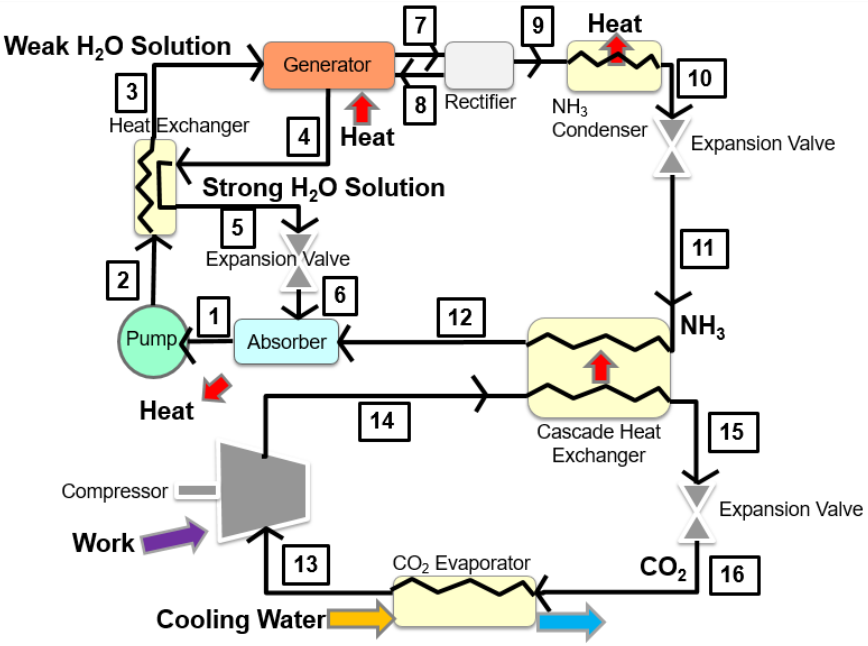

Figure 10: CO_2 – H_2O-NH_3 cascading compression and absorption refrigeration cycle diagram

All the cycles were analyzed in Engineering Equation Solver (EES), which provided state values for temperature, pressure, enthalpy, entropy, and quality. It also allowed the relationships and energy balances of different states to be input so that energy and mass flows could be determined. For the compression cycles, the pressure-enthalpy state diagrams were also produced. Using this software, different parameters could be varied so that more optimal input temperature values could be used that increased the coefficient of performance (COP). As a reference, a typical domestic refrigerator with a cooling capacity of around 200 watts has a COP of about 2.75, but based on temperature restrictions and larger cooling scale required for this application, the COPs for the following cycles are expected to be lower than this [28].

For all cycles analyzed, it was assumed that compressors had an efficiency of 85%, heat exchangers had an effectiveness of 80%, all expansion across valves were modeled as throttling processes, the amount of work required for any pumps was negligible, there are no pressure drops across any heat exchangers, and kinetic and potential energy effects were negligible.

4.2. CO_2 – CO_2 System

Table 2: CO_2 - CO_2 system energy and performance results

	Bottom Cycle	Top Cycle	Total
Work In (MW)	2.80	7.20	10.0
Flow Rate (kg/s)	62.74	90.65	-
Available Cooling (MW)	14.86	-	14.86
C.O.P.	-	-	1.49

The CO_2-CO_2 cycle was designed within the constraints listed in the Methodology section to produce the results shown in Table 2. The state values can be visualized better in the pressure-enthalpy diagram, shown in Figure 11. The input energy to the system was limited by the 10 MW work constraint, and it was split up so that most of the work was performed by the high temperature cycle, with the low temperature cycle requiring 2.80 MW and the high temperature cycle requiring 7.20 MW. Both cycles' evaporation temperatures were determined by performing parametric studies with each temperature serving as the independent variables, while total COP and mass flow rate of both cycles were the dependent variables. The temperature for the CO_2 entering the low temperature evaporator was chosen to be -30°C, and the temperature of the CO_2 exiting the high temperature side of the cascading heat exchanger was 15°C, which maximized COP while keeping flow rate for CO_2 below 100 kg/s. See Appendix A1: Parametric Study Results for CO2 – CO2 System for

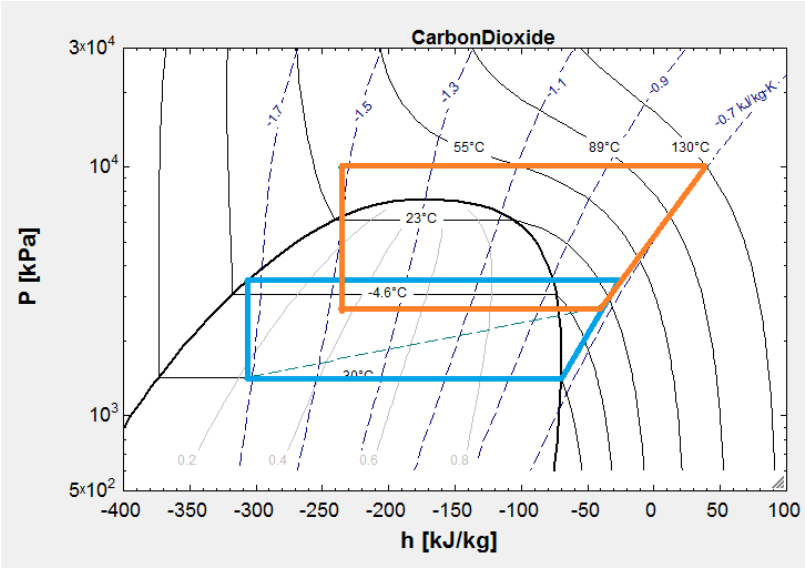

Figure 11: CO_2 - CO_2 state diagram on pressure vs. enthalpy chart with the blue cycle representing the low temperature side and the orange cycle representing the high temperature side

a visualization of the study. Finally, this cycle can provide a 14.86 MW cooling load to the cooling water. For all state values of the cycle, see Appendix A2: CO2 – CO2 Cycle Data.

4.3. CO_2 – NH_3 System

Table 3: CO_2 - NH_3 system energy and performance results

	CO_2 Cycle	NH_3 Cycle	Total
Work In (MW)	4.21	5.79	10.0
Flow Rate (kg/s)	94.39	24.17	-
Available Cooling (MW)	22.35	-	22.35
C.O.P.	-	-	2.24

The CO_2-NH_3 cycle was designed within the constraints listed in the Methodology section to produce the results shown in Table 3. The pressure-enthalpy diagram of the state values is shown in Figure 12.The input energy to the system was limited by the 10 MW work constraint, and it was split up relatively evenly between the cycles, with the low temperature cycle requiring 4.21 MW and the high temperature cycle requiring 5.79 MW. The CO_2 cycle's temperature entering the evaporator and the NH_3 cycle's pressure exiting the compressor were determined by performing parametric studies with both parameters serving as the independent variables, while total COP and mass flow rate of both

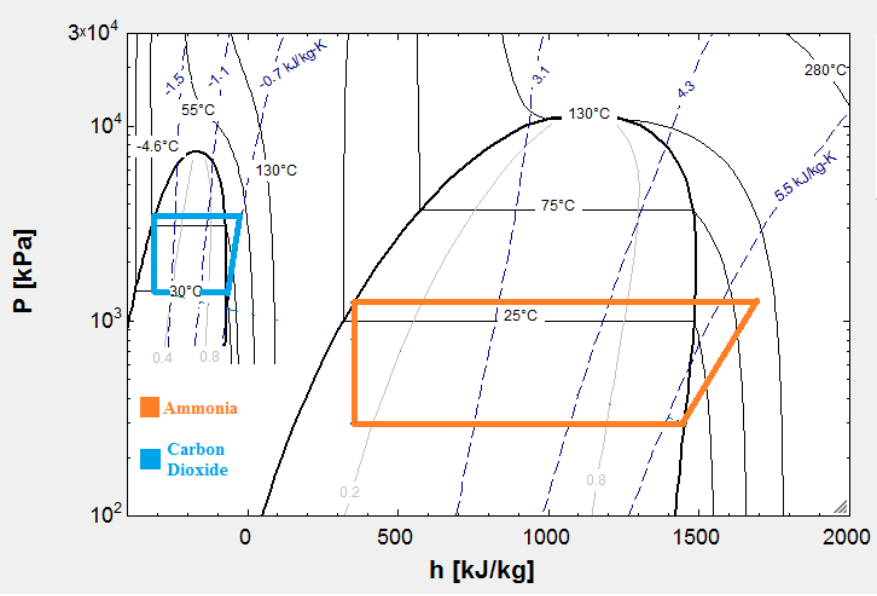

Figure 12: CO_2 – NH_3 state diagram on pressure vs. enthalpy chart with the blue cycle representing the low temperature side and the orange cycle representing the high temperature side

cycles were the dependent variables. The chosen temperature for CO_2 was -30°C and the chosen pressure for NH_3 was 1,250 kPa. These maximized COP while keeping flow rate for CO_2 below 100 kg/s. The results from the parametric study can be visualized in Appendix B1: Parametric Study Results for CO2 – NH3 System. Finally, this cycle can provide an 22.35 MW cooling load to the cooling water. For all state values of the cycle, see Appendix B2: CO2 – NH3 Cycle Data.

4.4. CO_2 – LiBr-H_2O System

Table 4: CO_2 - LiBr-H_2O system energy and performance results

	CO_2 Cycle	LiBr – H_2O Cycle	Total
Work In (MW)	8.04	<0.001	8.04
Heat In (MW)	0	30.0	30.0
Max. Flow Rate (kg/s)	100.0	83.8	-
Available Cooling (MW)	20.96	-	20.96
C.O.P.	-	-	0.762

The CO_2 – LiBr-H_2O cycle was also designed within the constraints and assumptions listed for the cycles, but the water in the absorption cycle was required to have a temperature in the evaporator greater than 0°C. This cycle produced the results shown in Table 4. This system's input energy was limited not by work but by heat to the absorption cycle, which was capped at 30 MW. The compression cycle required 8.04 MW. A pressure-enthalpy diagram for this system is less applicable, because the absorption cycle gains and releases energy mainly based on the LiBr-H_2O solution's mass makeup, not its pressure and enthalpy. For this cycle, the evaporation temperature of the absorption cycle was limited to 5°C, although it ended up being 7°C. The generator temperature was set to be 90°C. The temperature of the CO_2 entering the evaporator and the temperature of the CO_2 leaving the cascade heat exchanger were varied in a parametric

study like the ones performed in the first two cycles; the CO_2 entering the evaporator was chosen to be -40°C and the CO_2 leaving the heat exchanger was set to 10°C. The results from the study are illustrated in Appendix C1: Parametric Study Results from CO2 – LiBr-H2O System. The available cooling from the system was determined to be 20.9 MW. For all state values of the cycle, see Appendix C2: CO2 – LiBr-H2O Cycle Data.

4.5. $CO_2 – H_2O$-NH_3 System

Table 5: CO_2 - H_2O-NH_3 system energy and performance results

	CO_2 Cycle	$H_2O – NH_3$ Cycle	Total
Work In (MW)	1.00	<0.001	1.00
Heat In (MW)	0.0	30.0	30.0
Max. Flow Rate (kg/s)	55.89	51.06	-
Available Cooling (MW)	13.91	-	13.91
C.O.P.	-	-	0.449

The $CO_2 – H_2O$-NH_3 cycle was also designed within the constraints and assumptions listed for the cycles, but the ammonia did not have the same constraint as the water in the previous system, as its freezing point is -77.7°C. This cycle produced the results shown in Table 5. This system's input energy was limited not by work but by heat to the absorption cycle, which was capped at 30 MW. The compression cycle required only 1 MW of work. As noted for the previous system, a pressure-enthalpy diagram is less applicable here. For the absorption cycle, the temperature of the ammonia exiting the cascading heat exchanger was set to -12°C and the temperature of the carbon dioxide entering the low temperature evaporator was set to -20°C, but the for this system, the temperature of the weak solution leaving the absorber and the amount of work required for the compressor were varied in a parametric study. The dependent variable was the cooling load available to act on the cooling water. The results from this study are shown in

Appendix D1: Parametric Study Results for CO2 – H2O-NH3. The available cooling from the system was determined to be 13.9 MW. For all state values of the cycle, see Appendix D2: CO2 – H2O-NH3 Cycle Data.

4.6. Baseline Flow Rate

After determining the cooling loads available from each cycle and cooling water temperature requirements for each region, the baseline water use should be determined for each situation. The baseline case is simply the amount of water needed to cool the hot working water in the condenser. Based on EPRI standards, the hot working water comes in to the condenser as steam at 50℃ and leaves as a saturated liquid, also at 50℃, with a flow rate of 315 kg/s. Based on the enthalpies of the water at those two states, the required cooling was determined to be 750.6 MW. The equation for this is shown in Equation 1.

$$\dot{m}_{hot} \times (h_{out} - h_{in}) = Q_{cond} = 315\frac{kg}{s} \times \left(2592.1\frac{kJ}{kg} - 209.3\frac{kJ}{kg}\right) = 750.6\ MW \quad (1)$$

The average cooling water and ambient air temperatures for each region, determined from NCDC and USGS data as discussed in Section 3.4, were used to determine the properties of the cooling water and various temperature constraints in the cooling systems. These temperatures are shown in Table 6.

Table 6: Estimated average surface water and ambient air temperatures for each region

	Northeast	Midwest	Southeast	West
Water Temperature (℃)	12.8	19.5	24.0	15.8
Air Temperature (℃)	11.9	11.4	22.6	20.4

The once-through cooling systems will use surface water temperature as the inlet temperature and a ΔT of 10°C. The recirculating systems are a bit more complex. The outlet temperature from the cooling tower will be 5°C, the approach differential, higher than the incoming ambient air; it is assumed the cooling tower provides 10°C of cooling, so the outlet temperature of the cooling water from the condenser is 15°C higher than the ambient air. Finally, the inlet temperature is a weighted fraction of the temperature of the water out of the cooling tower and the makeup water temperature. It is assumed that the makeup water required is about 7.5% of the total flow of the water, and the water from the tower is 92.5%. For the location of the various temperatures, see Figure 13. The equations and sample calculations for calculating the inlet and outlet temperatures for recirculating systems are shown in Equations 2 and 3.

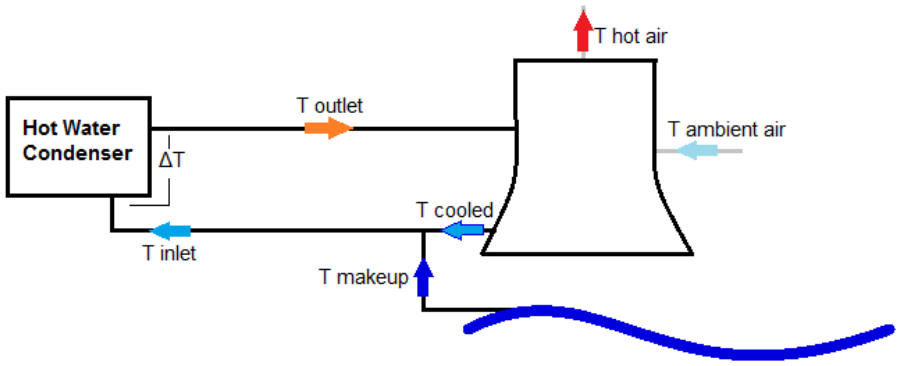

Figure 13: Location of temperatures and flows of recirculating cooling water and outside air

$$T_{out} = T_{air} + \Delta T_{cond} + \Delta T_{appr} = 11.9°C + 10°C + 5°C = 26.9°C \quad (2)$$

$$T_{in} = T_{mkup} \times \left(\frac{\dot{m}_{mkup}}{\dot{m}_{total}}\right) + T_{out} \times \left(\frac{\dot{m}_{out}}{\dot{m}_{total}}\right) = 11.2°C \times 7.5\% + 26.9°C \times 92.5\% = 26.9°C \quad (3)$$

The water properties were established based on the water outlet temperature using the NIST fluid property calculator [29]. The results of baseline flow are shown in Table 7.

Table 7: Baseline flow and parameter results based on region water and air properties and cooling type

	Northeast		Midwest		Southeast		West	
	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating
Inlet Temp. (°C)	11.2	16.5	17.3	16.5	24.0	27.3	15.8	24.7
Air Temp. (°C)	11.9		11.4		22.6		20.4	
Outlet Temp. (°C)	21.2	26.9	27.3	26.4	34.0	37.6	25.8	35.4
Water Density (kg/m^3)	997.98	996.59	996.48	996.70	994.40	993.13	996.86	993.92
Water Specific Heat (kJ/kg K)	4.183	4.181	4.180	4.181	4.182	4.179	4.181	4.179
ΔT (°C)	10.00	10.43	10.00	9.93	10.00	10.27	10.00	10.72
Baseline Flow Rate (kg/s)	**17,944**	**17,219**	**17,955**	**18,076**	**17,950**	**17,489**	**17,953**	**16,754**

4.7. Water Savings

Now that the baseline flow has been established, the total water savings for each kind of plant and refrigeration strategy can be determined. Regionally, the water savings will differ slightly because the properties of water, like density and heat capacity, vary at different temperatures. The water savings from the refrigeration cycles and using surface water as the inlet or makeup water are shown in Table 8, Table 9, Table 10, and Table 11. The water flow rate savings are calculated by using Equation 4, and sample calculations are shown; the makeup water savings for recirculating systems are calculated using Equation 5, which was developed in Perry's Chemical Engineers Handbook [22].

$$\dot{m}_{svd} = \frac{Q_{refrg}}{\Delta T \times c_p} = \frac{14.86\,MW}{10°C \times 4.1831\,\frac{kJ}{kg}} = 355.2\frac{kg}{s} \tag{4}$$

$$\dot{m}_{cons\_svd} = \left((55.08) \times \frac{\dot{m}_{svd}}{\rho}\right) \times 1.25 + (7.2) \times \frac{\dot{m}_{svd}}{\rho} \tag{5}$$

As mentioned in earlier sections, the relevant water savings for once-through systems are the flow savings, which correspond to a reduction in water withdrawal, whereas the relevant water savings for recirculating systems are the makeup water savings, which corresponds to consumption savings.

Table 8: Flow savings with the addition of a CO_2-CO_2 refrigeration cycle for each region and cooling type

	Northeast		Midwest		Southeast		West	
	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating
Baseline Flow Rate (kg/s)	17,953	17,228	17,965	18,086	17,960	17,498	17,963	16,763
Flow Rate with Refrigeration (kg/s)	17,598	16,887	17,609	17,728	17,604	17,152	17,607	16,432
Flow Savings (kg/s)	**355.2**	340.9	**355.5**	357.9	**355.4**	346.2	**355.4**	331.7
Makeup Water Savings (kg/s)	-	**7.23**	-	**7.58**	-	**7.36**	-	**7.05**
Percent of Total Saved	**2.0%**							

Table 9: Flow savings with the addition of a CO_2-NH_3 refrigeration cycle for each region and cooling type

	Northeast		Midwest		Southeast		West	
	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating
Baseline Flow Rate (kg/s)	17,953	17,228	17,965	18,086	17,960	17,498	17,963	16,763
Flow Rate with Refrigeration (kg/s)	17,419	16,715	17,430	17,548	17,425	16,977	17,428	16,265
Flow Savings (kg/s)	**534.3**	512.7	**534.6**	538.2	**534.5**	520.7	**534.6**	498.9
Makeup Water Savings (kg/s)	-	**10.9**	-	**11.4**	-	**11.1**	-	**10.6**
Percent of Total Saved	**3.0%**							

Table 10: Flow savings with the addition of a CO_2 - LiBr-H_2O refrigeration cycle for each region and cooling type

	Northeast		Midwest		Southeast		West	
	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating
Baseline Flow Rate (kg/s)	17,953	17,228	17,965	18,086	17,960	17,498	17,963	16,763
Flow Rate with Refrigeration (kg/s)	17,452	16,747	17,463	17,581	17,458	17,010	17,461	16,296
Flow Savings (kg/s)	**501.1**	480.8	501.4	504.8	**501.2**	488.4	**501.3**	467.9
Makeup Water Savings (kg/s)	-	**10.2**	-	**10.7**	-	**10.4**	-	**9.9**
Percent of Total Saved	**2.8%**							

Table 11: Flow savings with the addition of a CO_2 – H_2O-NH_3 refrigeration cycle for each region and cooling type

	Northeast		Midwest		Southeast		West	
	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating
Baseline Flow Rate (kg/s)	17,953	17,228	17,965	18,086	17,960	17,498	17,963	16,763
Flow Rate with Refrigeration (kg/s)	17,621	16,909	17,632	17,751	17,627	17,174	17,630	16,453
Flow Savings (kg/s)	**332.5**	319.1	**332.7**	335.0	**332.6**	324.1	**332.7**	310.5
Makeup Water Savings (kg/s)	-	**6.8**	-	**7.1**	-	**6.9**	-	**6.6**
Percent of Total Saved	**1.9%**							

4.8. Costs and Benefits of Refrigeration Cycle Recommendations

With the savings for all the refrigeration cycles and water sources determined, the final step is to determine the most appropriate water reduction strategy for each region and cooling type and determine the economic benefits and costs of the strategy. The annual economic

benefits of the cycle are determined by applying an estimate for the cost of water to power plants developed by Ulrich, et al [23], although, as stated earlier, economic savings cannot be appropriately calculated for recirculating systems that consume water as opposed to withdrawing it. This equation is shown below in Equation 6.

$$cost_{water}, \frac{\$}{\frac{kg}{s}} = \left[\left(0.0001 + 3 \times 10^{-5} \times \left(\frac{\dot{m}_{c.w.}}{\rho}\right)\right) \times CPI + 0.003 \times cost_{fuel}\right] \times \rho \quad (6)$$

The annual energy costs of the cycles were determined by using the average price of industrial electricity for regions designated by EIA that were the most like regions used in this study. These prices can be seen in Table 12 [30].

Table 12: Estimated average price of industrial electricity based on region

Region	Northeast	Midwest	Southeast	West
Price of Electricity ($/kWh)	$0.0688	$0.0704	$0.0691	$0.0822

The annual energy costs were assumed to be the annual energy use of the compressors multiplied by the price of electricity. The energy for heat in the absorption cycles was assumed to cost nothing, as it could be captured as waste heat from a typical power plant. This is not entirely true, as there would have to some energy put into making the heat usable, but compared to the compression work, it is assumed to be negligible. The annual energy of each cycle is shown in Table 13, and the annual energy cost will be determined for the cooling strategy used in each region. The capital costs of the refrigeration cycles were assumed to be equal to the costs of the compressors required by the systems because compressors would be by far the most expensive part of a refrigeration cycle. The cost of the compressors was determined by a compressor cost rating system developed by Amin Almasi [31]; the costs, outlined in Table 13,

are just estimates but can provide a good idea of the magnitude of savings necessary for the system to be economically viable.

Table 13: Estimated annual energy use and capital cost of refrigeration cycles

	CO_2-CO_2	CO_2-NH_3	CO_2 - LiBr-H_2O	CO_2 – H_2O-NH_3
Annual Energy Use (kWh/year)	87,600,000	87,600,000	64,272,120	8,760,000
Capital Cost ($)	$14,650,000	$14,380,000	$9,692,778	$2,152,778

For the Northeast region, the refrigeration cycle recommended was the CO_2 – H_2O-NH_3 cycle, because the region does not experience a lot of water stress, so it does not make sense to spend a lot of money, annually and upfront, to reduce cooling water use. For the Midwest region, the refrigeration cycle recommended was the CO_2 – LiBr-H_2O cycle, because the region does experience some water stress along its major rivers like the Ohio and Mississippi. For the Southeast region, the refrigeration cycle recommended was also the CO_2 – LiBr-H_2O cycle, because the region experiences some water stress along the Mississippi, but, more importantly, the region struggles with cooling water return temperature for the once-through systems, so in exchange for some water reduction, the return temperature could be reduced using the refrigeration cycle. Still, it is not paramount for the cycle to have the maximum reduction of water use in exchange for high costs, so the CO_2 – NH_3 cycle, which has the highest water reduction potential, was not recommended. For the West region, the refrigeration cycle recommended was the CO_2 – NH_3 cycle, because the region experiences extreme water stress in many different areas that have power plants. There are power plants that this may not apply to in the West region, but in the West more than any region, maximum water savings are paramount. The water savings, cost savings, and energy costs for each recommendation are shown in Table 14. The equations used to calculate water saved and energy cost, along with sample calculations, are shown in Equations 7 and 8.

$$Water\ Saved_{annual} = \frac{\dot{m}_{svd}}{\rho} \times 3{,}600\,\frac{s}{hr} \times 8{,}760\,\frac{hr}{yr} \times 264.17\,\frac{gal}{m^3} = 2.78 \times 10^9\,\frac{gal}{year} \quad (7)$$

$$Energy\ Cost_{annual} = W_{in} \times 8{,}760\,\frac{hr}{year} \times Cost_{elec} = 10{,}000\,kW \times 8{,}760\,\frac{hr}{year} \times \frac{\$0.0688}{kW} = \frac{\$6{,}026{,}880}{year} \quad (8)$$

Table 14: Water savings and economics costs & benefits for cooling strategy recommendations

Region	Northeast		Midwest		Southeast		West	
Recommended Cycle	CO2-H2O-NH3		CO2-LiBr-H2O		CO2-LiBr-H2O		CO2-NH3	
Cooling Type	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating
Water Saved (Million gal/year)	2,776	56.5	4,192	89.4	4,199	87.1	4,467	88.9
Annual Water Cost Savings ($/year)	$787,394	$16,047	$1,188,990	$25,363	$1,191,108	$24,724	$1,267,213	$25,231
Annual Energy Cost ($/year)	$602,688		$4,524,757		$4,441,203		$7,200,720	

4.9. Potential Savings from Using Groundwater

An additional strategy that was analyzed after obtaining water savings results from the refrigeration cycles was using groundwater or aquifer water as cooling water instead of surface water, which most power plants use. Determined using the methodology described in Section 3.4, the water temperatures for both surface and aquifer sources of the four different regions are shown in Figure 14 [12, 21]. It was hypothesized that the aquifer water would be cooler than the surface water, and this was true in all regions except the West region. These values

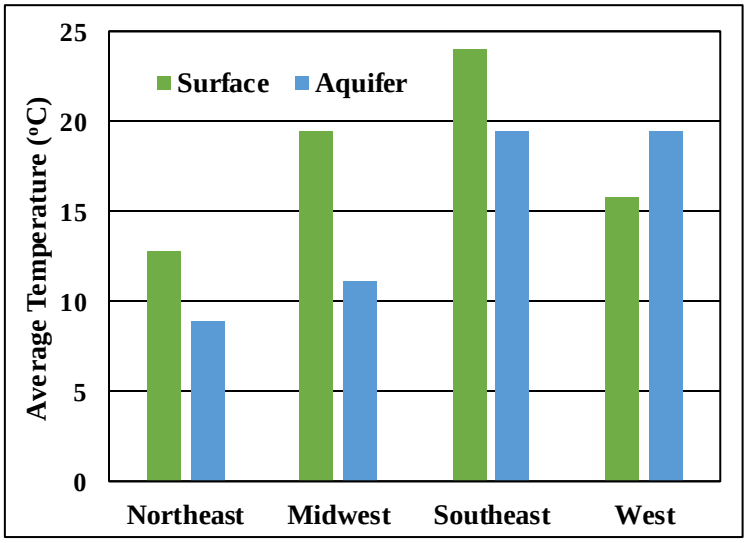

Figure 14: Average water temperatures for different regions and sources

should be used cautiously, as temperatures vary widely from source to source, and the regions used each cover many different climate zones. However, they should serve as a good comparison tool.

The savings that are associated with using groundwater, especially for the once-through systems, should be treated as rough theoretical estimates, as using ground water in these quantities may not be possible, or it may deplete desperately needed drinking water sources in some areas. In fact, the amount of water that would be required for once-through systems is most likely an unreasonable strain on groundwater sources that are used for other purposes. For this reason, the percent savings for once-through systems are not shown, so as not to present potentially misleading results. The results for flow and makeup water savings are shown in Table 15.

Table 15: Water flow savings created from using groundwater instead of surface water for cooling

	Northeast		Midwest		Southeast		West	
	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating	Once-through	Recirculating
Inlet Temp. (°C)	8.9	16.3	11.1	16.0	19.4	27.0	19.4	25.0
Air Temp. (°C)	11.9		11.4		22.6		20.4	
Outlet Temp. (°C)	21.2	26.9	27.3	26.4	34.0	37.6	25.8	35.4
Water Density (kg/m^3)	997.98	996.59	996.48	996.7	994.4	993.13	996.86	993.9
Water Specific Heat (kJ/kg K)	4.183	4.181	4.180	4.181	4.182	4.179	4.181	4.17
ΔT (°C)	12.30	10.60	16.20	10.40	14.60	10.62	6.40	10.4
Baseline Flow Rate (kg/s)	17,944	17,219	17,955	18,076	17,950	17,489	17,953	16,75
Flow Rate Using Groundwater (kg/s)	14,588	16,938	11,083	17,268	12,295	16,920	28,052	17,18
Flow Savings (kg/s)	**3,355**	280	**6,872**	808	**5,656**	568	**-10,099**	-43
Makeup Water Savings (kg/s)	-	**5.9**	-	**17.1**	-	**12.1**	-	**-9.2**
Percent of Total Saved	-	**1.6%**	-	**4.5%**	-	**3.3%**	-	**-2.6%**

5. Summary and Conclusions

The goal of this study was to design and evaluate strategies for reducing cooling water use in power plants. For recirculating plants, this meant reducing water consumption, and for once-through plants, this meant reducing water withdrawal. To evaluate the effectiveness of the strategies that were developed, four regions were established and water and air temperatures were estimated for the regions. To reduce the cooling water use, the temperature differential across the hot water condenser needed to be increased. To allow for this without increasing the outlet temperature, it was necessary to design refrigeration cycles to reduce the temperature of

cooling water after the hot water condenser. Cycles were designed in accordance with assumptions and constraints that were developed based on typical power plant properties, refrigerant qualities, and water and air temperatures, among other things. After the cycles were designed and water temperatures established, the flow reduction, and thus water savings, could be established. A strategy for each region and cooling type was suggested based on the water savings and the different water landscapes in the various regions. To provide an idea of how economically feasible the strategies were, the water saving cost benefits, energy costs, and cycle capital costs were all estimated. For the sake of investigating strategies that could be coupled with the refrigeration cycles, different water sources were also evaluated to determine if drawing water from those sources could help reduce cooling water use based on the temperature of the other sources.

The most important metric gained from this research is the potential amount of water that could be saved. The energy-water nexus has become one of the largest issue surrounding power plants in recent years, and there has been a call for solutions to help reduce the water use in power plants. There will not be one simple solution that solves the problem; it will be solved iteratively, with unique solutions implemented at different power plants. That is why the economic results of this research helpful for comparison, but are not the best way to judge feasibility or significance. There is neither a high confidence in their accuracy or precision, nor does the economic cost and benefit capture all of the factors involved in how the reduction of cooling water will affect the power plant and its surrounding region. With that said, a single once-through plant could see water cost savings of around $1,000,000 per year, which is perhaps marginal compared to the power plant profits, but cannot be ignored in terms of magnitude. For recirculating plants, the savings are smaller at around $25,000 per year, but this does not capture

the benefits of keeping water in the local watershed that would otherwise have been lost to evaporation or another form of consumption. Also, as populations and economies continue to grow, the price of water will continue to rise due to increased scarcity, along with increased demand for agriculture and energy production. So, while these strategies may not be economically beneficial in current economic times, it is easy to see how water will start to become the U.S.'s most valuable and expensive resources, especially in certain regions like California.

In terms of specific results, the water savings for these cycles is significant. Regardless of the region or plant type, using these refrigeration cycles could result in saving two to three percent of all water used. While this may not sound like a huge reduction, this corresponds to around 4 billion gallons of water per year saved from withdrawal at a single once-through plant and 80 million gallons of water per year saved from consumption at a single recirculating plant. This is a huge amount of water that would be newly available to the surrounding areas for drinking water, industrial use, or ecological processes. Specifically, the ability to save tens of millions of gallons of water from consumption at a single recirculating plant is extremely significant for areas that experience a lot of water stress, like the Southwest, the Mississippi River region, and the eastern part of Pacific Coast states. Currently, economic metrics do not appropriately capture the true value of water in these regions, but the value of water will continue to rise in coming years, as the rate of water use continues to increase, while water availability remains the same or decreases. There is even potential for power plants to trade their water savings, as some industries trade electricity savings, although the infrastructure for this kind of system would require more research.

The water savings for once-through systems is also important, although for different reasons. In many water ecosystems, the rising temperature of the water has decreased the oxygen availability in the water and has ripened environments for harmful algal blooms, as mentioned in the introduction. If the thermal mass added to these rivers from high-temperature cooling water can be decreased, either by a reduction in flow or a reduction in temperature, which are both possible through the refrigeration cycles, these environmentally and ecologically harmful events can be slowed, if not eliminated.

It is also important to note that most of the systems chosen use heat as an input for the absorption cycle. This was treated as "free" energy in this study due to the ability to capture a certain amount of usable waste heat from the power plant, but there is significant infrastructure that must be implemented to capture this heat, and it may not be available for all power plants.

This reasoning goes for the use of aquifer water as well. The savings results for using groundwater instead of surface water at recirculating plants, although not at once-through plants due to the unreasonable stress they would put on an aquifer, are just as significant as, if not more than, the potential savings from the refrigeration cycles. In the Midwest and Southeast, savings could be as high as 4% of total cooling water consumption. However, there are concerns with the impact of extracting large amounts of ground water from an aquifer, and along with that also comes economic and logistical issues about the infrastructure that would need to be implemented. Nonetheless, neither of these issues are new to the water or energy industry, and existing solutions can most likely be adapted to fit the cases presented in this research. Future research into the effects of using significant amounts of groundwater, on the order of billions of gallons per year, for power plant cooling could reveal another strategy for reducing the amount cooling water use at power plants.

The impact of this research is evident in the water savings that are shown to be possible, but there are many steps that must be taken before these strategies can be put in place. Now that the cycles have been theoretically designed, the ones that were selected for use in different regions need to be designed at a lab scale to assure that they provide the expected cooling based on the scaled-down size they were designed at. Next, specific power plants should be chosen as case studies to determine logistically the best way to lay out these refrigeration systems and, where applicable, the groundwater extraction. This includes how to harness waste heat and how to source energy for the compressors. The groundwater extraction would also likely require environmental surveys and governmental permission. Ultimately, for this research to be used as motivation for power plants to install these refrigeration cycles, the true value of water for different regions will have to be quantified or at least understood well enough to weight the costs and benefits of these strategies.

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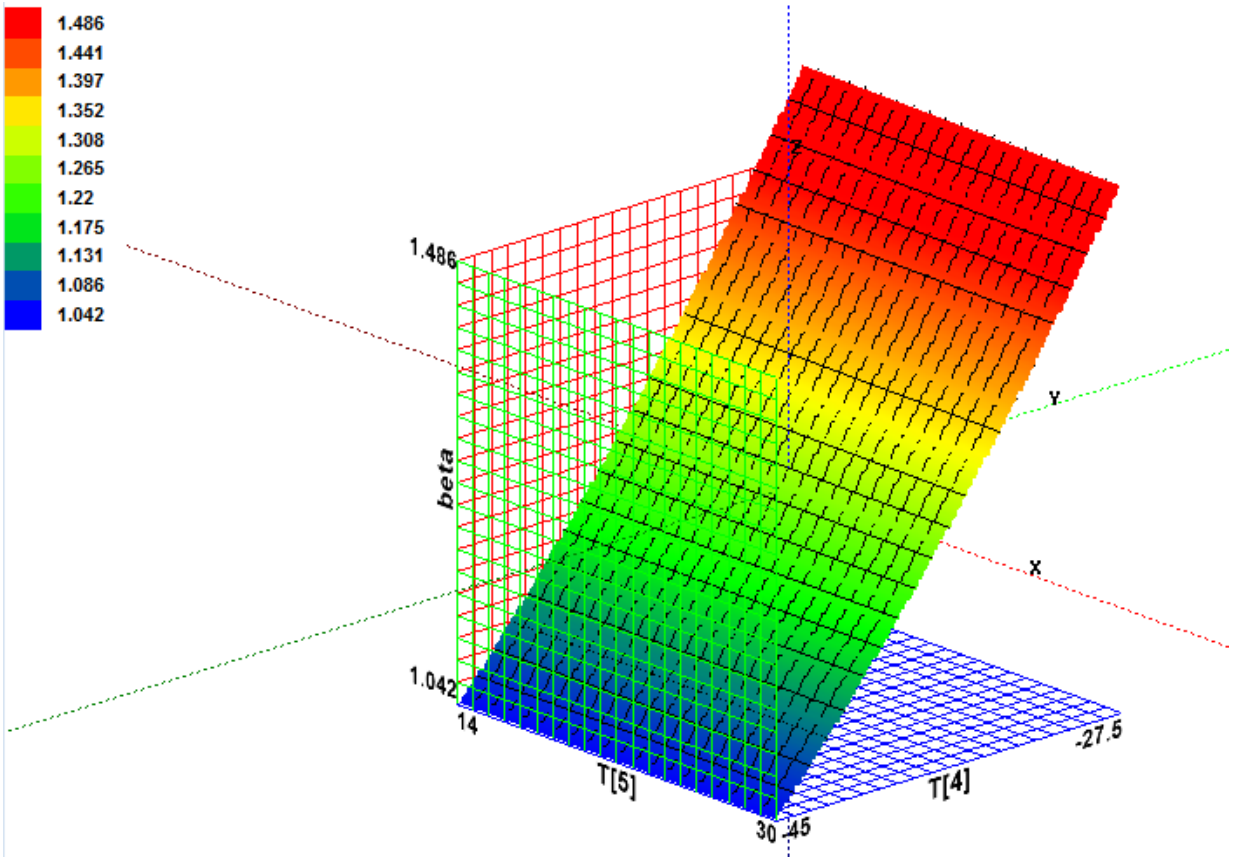
Appendix A1: Parametric Study Results for CO_2 – CO_2 System



Appendix A2: CO_2 – CO_2 Cycle Data

Solution

Main

Unit Settings: SI C kPa kJ mass deg

β = 1.486	δ_T = -9.004 [C]	η = 0.85	h2s = -32.06 [kJ/kg]
h6s = 27.15 [kJ/kg]	HxEf = 0.8	Q_{in} = 14858 [kW]	Q_{out} = 24858 [kW]
R$ = 'CarbonDioxide'	s2s = -0.741 [kJ/kg*K]	s6s = -0.7311 [kJ/kg*K]	

Arrays Table

Main

Sort	1 h_i [kJ/kg]	2 $\dot{m}_i$ [kg/s]	3 P_i [kPa]	4 s_i [kJ/kg*K]	5 T_i [C]	6 $\dot{W}_i$ [kW]	7 x_i
[1]	-69.96	62.74	1428	-0.741		2798	1
[2]	-25.37	90.65	3485		36.02	7202	
[3]	-306.8		3485		0		0
[4]	-306.8		1428		-30		
[5]	-40.39		2724	-0.7311	15		
[6]	39.07		10000				
[7]	-235.2		10000		30		
[8]	-235.2		2724		-9.004		

Appendix A3: CO_2 – CO_2 Cycle EES Code

```
"Refrigerant"
R$='CarbonDioxide'

"Assumptions"
W_dot[2]+W_dot[1]=10000 [kW]
eta=0.85 "Efficiency of compressors"
HxEf=0.8

"State 7"
P[7]=10000[kPa]
T[7]=30[C]
h[7]=enthalpy(R$,T=T[7],P=P[7])

"State 8"
h[8]=h[7]
{T[8]=-2.5}
P[8]=pressure(R$,h=h[8],T=T[8])

"State 5"
P[5]=P[8]
T[5]=15[C]
h[5]=enthalpy(R$,P=P[5],T=T[5])
s[5]=entropy(R$,P=P[5],T=T[5])

"State 6s"
s6s=s[5]
P[6]=P[7]
h6s=enthalpy(R$,s=s6s,P=P[6])

"State 6"
h[6]=(h6s-h[5])/eta+h[5]

"Heat Exchanger"
HxEf=(T[2]-T[3])/(T[2]-T[8])

"State 4"
T[4]=-30[C]
h[4]=h[3]
P[4]=pressure(R$,h=h[4],T=T[4])

"State 1"
P[1]=P[4]
x[1]=1
h[1]=enthalpy(R$,P=P[1],x=x[1])
s[1]=entropy(R$,P=P[1],x=x[1])

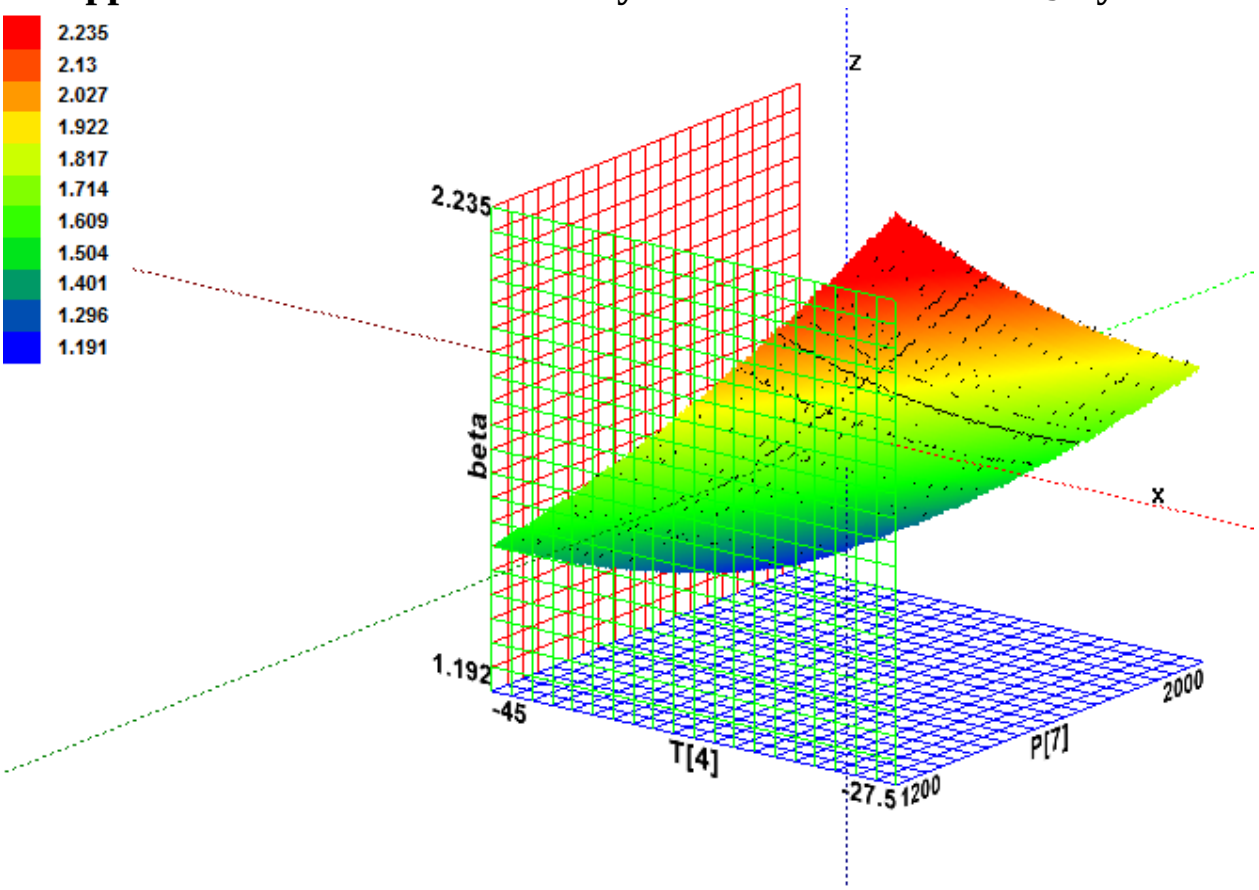
"State 2s"
s2s=s[1]
P[2]=P[3]
h2s=enthalpy(R$,s=s2s,P=P[2])
```

```
"State 2"
h[2]=(h2s-h[1])/eta+h[1]
T[2]=temperature(R$,h=h[2],P=P[2])

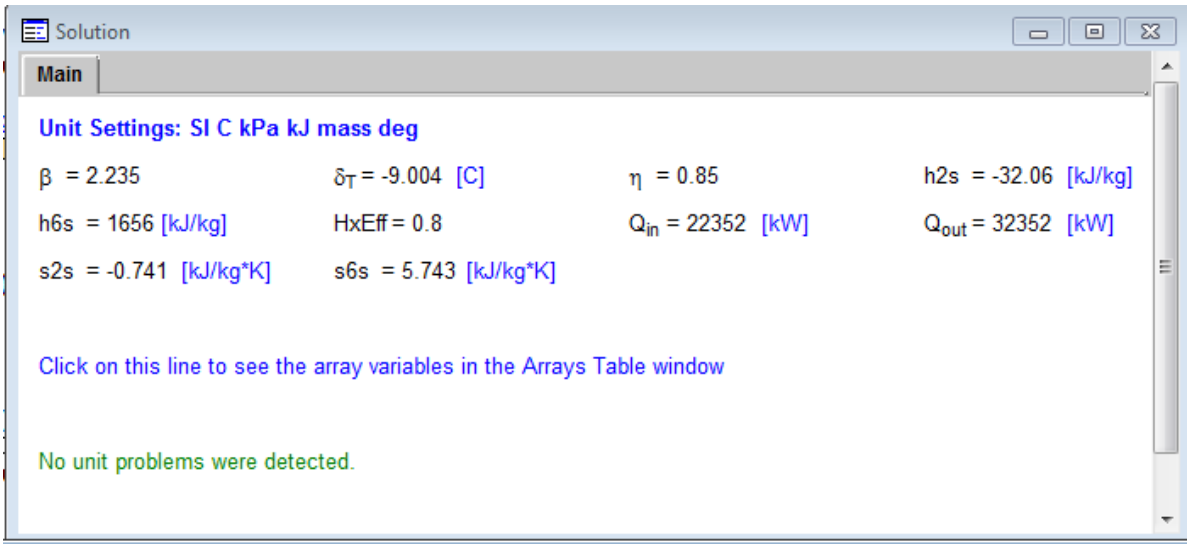
"State 3"
T[3]=0[C]
x[3]=0
P[3]=pressure(R$,x=x[3],T=T[3])
h[3]=enthalpy(R$,x=x[3],T=T[3])

"Equations"
m_dot[2]=W_dot[2]/(h[6]-h[5])
m_dot[1]=W_dot[1]/(h[2]-h[1])
(h[8]-h[5])*m_dot[2]=(h[3]-h[2])*m_dot[1]
Q_out=(h[6]-h[7])*m_dot[2]
Q_in=(h[1]-h[4])*m_dot[1]
beta=Q_in/(W_dot[1]+W_dot[2])
delta_T=T[8]-T[3]
```

Appendix B1: Parametric Study Results for CO_2 – NH_3 System



Appendix B2: CO_2 – NH_3 Cycle Data



Arrays Table

Main

Sort	1 h_i [kJ/kg]	2 $\dot{m}_i$ [kg/s]	3 P_i [kPa]	4 s_i [kJ/kg*K]	5 T_i [C]	6 $\dot{W}_i$ [kW]	7 x_i
[1]	-69.96	94.39	1428	-0.741	-30	4209	1
[2]	-25.37	24.17	3485		36.02	5791	
[3]	-306.8		3485		0		0
[4]	-306.8		1428		-30		
[5]	1452		302.8	5.743			1
[6]	1691		1250		106.4		
[7]	352.9		1250		32.32		0
[8]	352.9		302.8		-9.004		0.1503

Appendix B3: CO_2 – NH_3 Cycle Code

```
"Assumptions"
W_dot[2]+W_dot[1]=10000 [kW]
eta=0.85 "Efficiency of compressors"
HxEff=0.8

"State 7"
P[7]=1250[kPa]
x[7]=0
h[7]=enthalpy('Ammonia',x=x[7],P=P[7])
T[7]=temperature('Ammonia',x=x[7],P=P[7])

"State 8"
h[8]=h[7]
P[8]=pressure('Ammonia',h=h[8],T=T[8])
x[8]=quality('Ammonia',h=h[8],T=T[8])

"State 5"
P[5]=P[8]
x[5]=1
h[5]=enthalpy('Ammonia',P=P[5],x=x[5])
s[5]=entropy('Ammonia',P=P[5],x=x[5])

"State 6s"
s6s=s[5]
P[6]=P[7]
h6s=enthalpy('Ammonia',s=s6s,P=P[6])

"State 6"
h[6]=(h6s-h[5])/eta+h[5]
T[6]=temperature('Ammonia',P=P[6],h=h[6])

 "Heat Exchanger"
HxEff=(T[2]-T[3])/(T[2]-T[8])

"State 4"
T[4]=-30[C]
h[4]=h[3]
P[4]=pressure('CarbonDioxide' ,h=h[4],T=T[4])

"State 1"
P[1]=P[4]
x[1]=1
h[1]=enthalpy('CarbonDioxide' ,P=P[1],x=x[1])
s[1]=entropy('CarbonDioxide' ,P=P[1],x=x[1])
T[1]=temperature('CarbonDioxide' ,P=P[1],x=x[1])

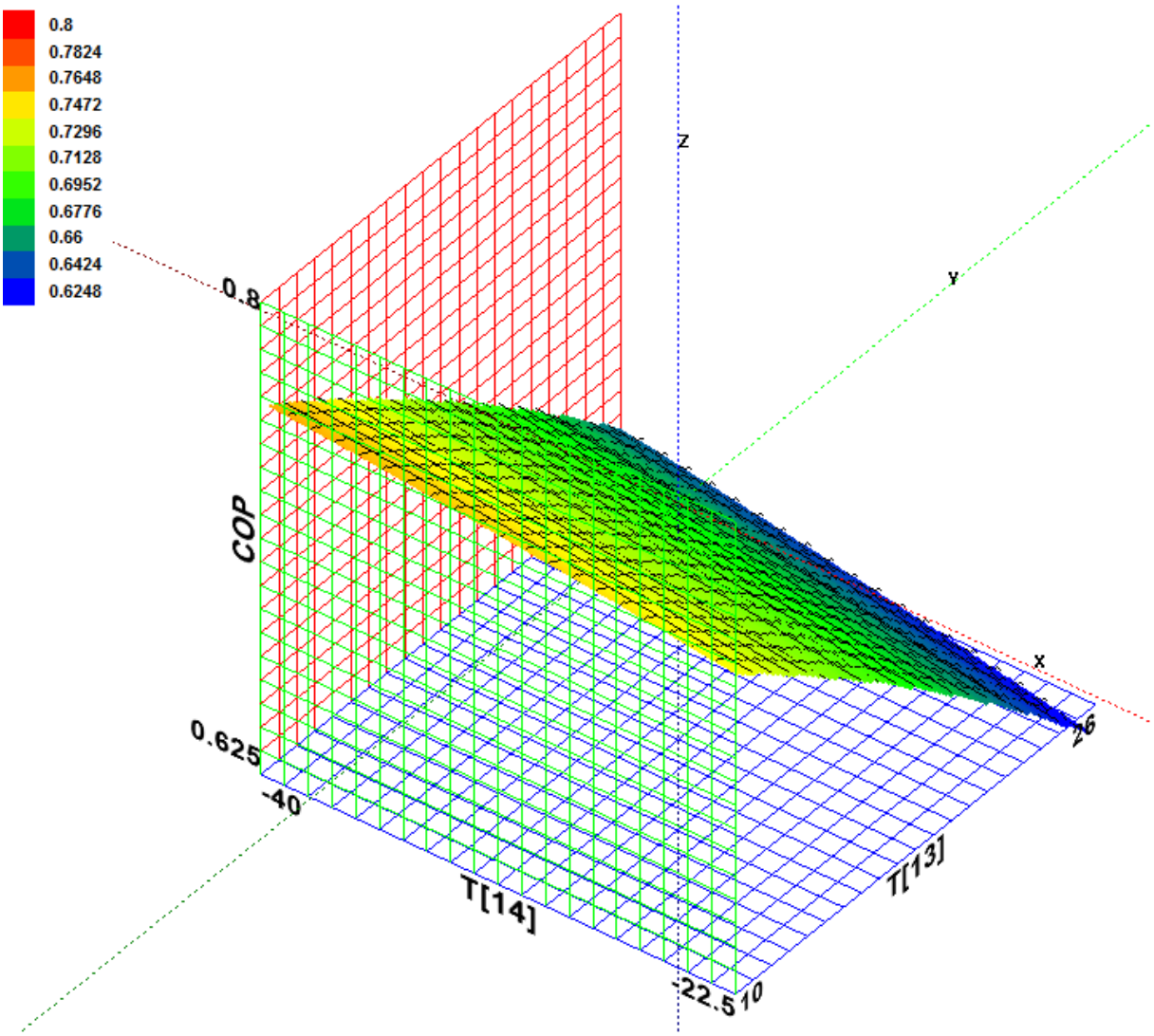
"State 2s"
s2s=s[1]
P[2]=P[3]
h2s=enthalpy('CarbonDioxide' ,s=s2s,P=P[2])
```

```
"State 2"
h[2]=(h2s-h[1])/eta+h[1]
T[2]=temperature('CarbonDioxide',P=P[2],h=h[2])

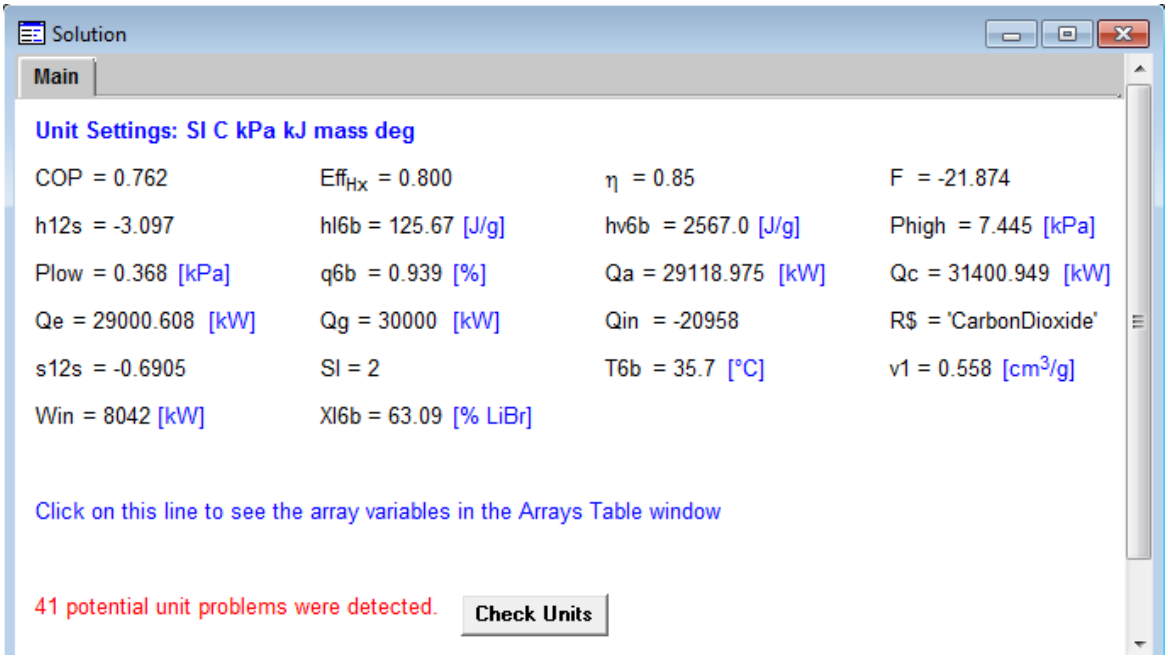
"State 3"
T[3]=0[C]
x[3]=0
P[3]=pressure('CarbonDioxide' ,x=x[3],T=T[3])
h[3]=enthalpy('CarbonDioxide' ,x=x[3],T=T[3])

"Equations"
m_dot[2]=W_dot[2]/(h[6]-h[5])
m_dot[1]=W_dot[1]/(h[2]-h[1])
(h[8]-h[5])*m_dot[2]=(h[3]-h[2])*m_dot[1]
Q_out=(h[6]-h[7])*m_dot[2]
Q_in=(h[1]-h[4])*m_dot[1]
beta=Q_in/(W_dot[1]+W_dot[2])
delta_T=T[8]-T[3]
```

Appendix C1: Parametric Study Results from CO_2 – LiBr-H_2O System



Appendix C2: CO_2 – LiBr-H_2O Cycle Data



Arrays Table

Main

Sort	1 h_i [mixed]	2 m_i [mixed]	3 P_i [kPa]	4 Q_i [Fraction]	5 T_i [mixed]	6 X_i [% LiBr]	7 s_i
[1]	148.1	-273.2811	0.368	0.000	40.0	65.4	
[2]	148.1	-273.2811	7.445		40.0	65.4	
[3]	220.3	-273.2811	7.445		80.5	65.4	
[4]	222.9	-285.7744	7.445	0.000	90.0	62.5	
[5]	148.6	-285.7744	7.445		50.0	62.5	
[6]	148.6	-285.7744	0.368	0.009	35.7	62.5	
[7]	2681.6	12.4934	7.445		97.0	0.0	
[8]	168.2	12.4934	7.445	0.000	40.2	0.0	
[9]	168.2	12.4934	0.368	0.181	-6.0	0.0	
[10]	2489.5	12.4934	0.368	1.000	-6.0	0.0	
[11]	-71.5	100.0000	1004.519	1.000	-40.0		-0.6905
[12]	9.0		4502.214		74.1		-0.6552
[13]	-281.0		4502.214	0.000	10.0		-1.651
[14]	-281.0		1004.519		-40.0		-1.589

Appendix C3: CO_2 – LiBr-H_2O Cycle Code

```
SI=2

{Input data}
Eff_Hx=0.8
Qg=30000
Phigh=7.445
T[4]=90
T[1]=40

Q[8]=0
Q[10]=1.0
x[4]=X_LIBR(T[4],Phigh,SI)
X[1]=X_LIBR(T[1],Plow,SI)

T[7]=T_LIBR(Phigh,x[3],SI)

{Set  pressures}
Phigh=pressure(WATER,T=T[8],x=Q[8])
Plow=pressure(WATER,T=T[10],x=Q[10])

{Heat Exchanger}
Eff_Hx=(T[4]-T[5])/(T[4]-T[2])
{Heat transfer}
                              {Energy balance}

{Generator}
m[3]=m[4]+m[7]
{Overall mass balance}
m[3]*x[3]=m[4]*x[4]
{LiBr balance}
h[3]*m[3]-h[4]*m[4]-h[7]*m[7]+Qg=0                          {Energy balance}

{Condenser}
Qc=m[7]*(h[7]-h[8])
{Energy balance}

{Refrigerant Valve}
h[9]=h[8]
{Energy balance}

{Evaporator}
Qe=m[9]*(h[10]-h[9])                                                {Energy
balance}

{Absorber}
m[10]*h[10]+h[6]*m[6]-Qa-m[1]*h[1]=0                       {Energy balance}

{Solution expansion valve model}
h[6]=h[5]
{Energy balance}
```

```
{Pump calculation}
h[2]=h[1]                              {Energy balance}

{Trivial mass balances}
m[2]=m[1]
m[3]=m[2]
m[5]=m[4]
m[6]=m[5]
m[8]=m[7]
m[9]=m[8]
m[10]=m[9]

x[2]=X[1]
x[3]=x[2]
x[5]=x[4]
x[6]=x[5]
x[7]=0
x[8]=x[7]
x[9]=x[8]
x[10]=x[9]

{Compute thermodynamic properties}
h[1]=H_LIBR(T[1],X[1],SI)
h[2]=H_LIBR(T[2],x[2],SI)
h[3]=H_LIBR(T[3],x[3],SI)
h[4]=H_LIBR(T[4],x[4],SI)
h[5]=H_LIBR(T[5],x[5],SI)
CALL Q_LIBR(h[6],Plow,x[6],SI:q6b,T6b,Xl6b,hl6b,hv6b)
T[6]=T6b
h[7]=enthalpy(WATER,T=T[7],P=Phigh)
h[8]=enthalpy(WATER,T=T[8],x=0)
T[9]=temperature(WATER,h=h[9],P=Plow)
h[10]=enthalpy(WATER,T=T[10],x=1)

v1=V_LIBR(T[1],X[1],SI)

{Compute COP}
COP=Qe/(Win+Qg)

{Solution Circulation Ratio}
F=x[4]/(x[4]-x[3])

{Vapor quality at 9}
q[9]=quality(STEAM_NBS,h=h[9],P=Plow)


{Set Pressures}
P[1]=Plow
P[2]=Phigh
P[3]=Phigh
P[4]=Phigh
P[5]=Phigh
```

```
P[6]=Plow
P[7]=Phigh
P[8]=Phigh
P[9]=Plow
P[10]=Plow

{Set Vapor Quality}
Q[1]=0
Q[4]=0
Q[6]=Q6b*0.01          {Fraction}
{Q[9]=quality(STEAM_NBS,h=h[9],P=Plow)}


"CO2 Cycle"
"Refrigerant"
R$='CarbonDioxide'
"State 14"
T[14]=-40[C]
h[14]=h[13]
P[14]=pressure(R$,h=h[14],T=T[14])
s[14]=entropy(R$,h=h[14],T=T[14])

"State 11"
P[11]=P[14]
Q[11]=1
h[11]=enthalpy(R$,P=P[11],x=Q[11])
s[11]=entropy(R$,P=P[11],x=Q[11])
T[11]=temperature(R$,P=P[11],h=h[11])

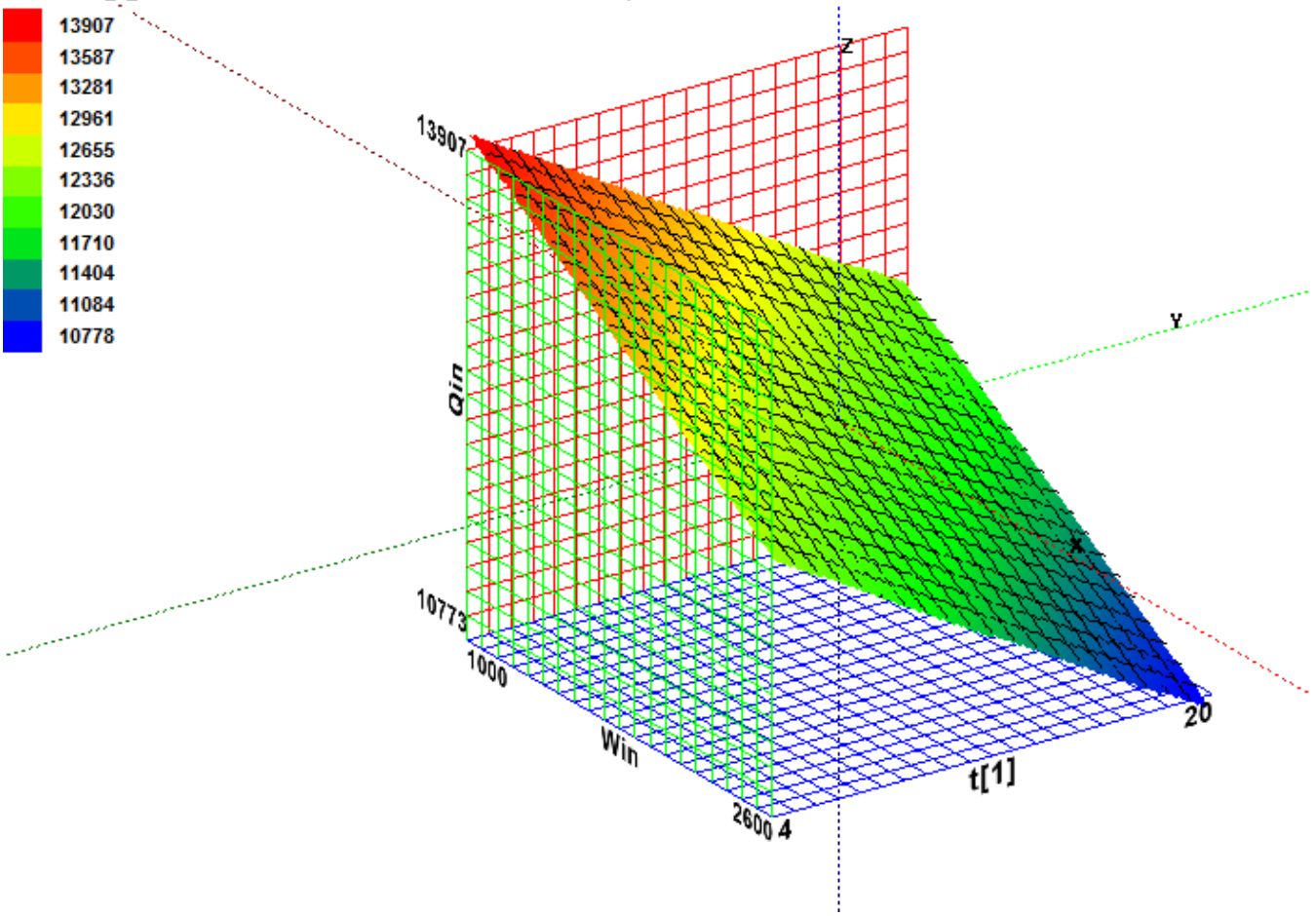
"State 12s"
s12s=s[11]
P[12]=P[13]
h12s=enthalpy(R$,s=s12s,P=P[12])

"State 12"
h[12]=(h12s-h[11])/eta+h[11]
T[12]=temperature(R$,P=P[12],h=h[12])
s[12]=entropy(R$,P=P[12],h=h[12])

"State 13"
Eff_Hx=(T[12]-T[13])/(T[12]-T[9])
T[13]=10[C]
Q[13]=0
P[13]=pressure(R$,x=Q[13],T=T[13])
h[13]=enthalpy(R$,x=Q[13],T=T[13])
s[13]=entropy(R$,P=P[13],h=h[13])

"Equations"
m[11]=100
eta=0.85 "Efficiency of compressor"
m[11]=Win/(h[12]-h[11])
(h[10]-h[9])*m[10]=(h[12]-h[13])*m[11]
Qin=(h[14]-h[11])*m[11]
```

Appendix D1: Parametric Study Results for $CO_2 - H_2O$-NH_3



Appendix D2: CO_2 – H_2O-NH_3 Cycle Data

Solution

Main | TPQ | TPX | PXQ | PHX | TXQ | SHX

Unit Settings: SI C kPa kJ mass deg

bala = -0.000000	balm = -0.000000	checkQ = 0.000004
COP = 0.449	Dx = 0.200	eshx = 0.800
η = 0.85	f = 3.876 [kg rich sol./kg vapor]	h16s = -54.68 [kJ/kg]
h3e = 859.415 [kJ/kgK]	h3f = 141.739 [kJ/kgK]	h5e = -128.695 [kJ/kgK]
phigh = 1166.917 [kPa]	plow = 88.949 [kPa]	pratio = 13.119
Q3e = 0.338	Q5e = -0.001	Qabs = 26706 [kW]
Qcond = 15431 [kW]	Qevap = 14907.2 [kW]	Qgen = 30000.0 [kW]
Qin = 13907 [kW]	Qrect = 2770.0 [kW]	Qshx = 16186.6 [kW]
R$ = 'CarbonDioxide'	s16s = -0.7982 [kJ/kgK]	s2s = -0.7905 [kJ/kgK]
s3e = 2.923 [kJ/kgK]	s3f = 1.038 [kJ/kgK]	s5e = 0.043 [kJ/kgK]
T3f = 83.724 [C]	tglide = 23.887 [C]	v3e = 0.054 [m^3/kg]
v3f = 0.001 [m^3/kg]	v5e = 0.001 [m^3/kg]	Win = 1000 [kW]
Wp = 0.0 [kW]		

Arrays Table

Main

Sort	1 h_i [kJ/kg]	2 m_i [kg/s]	3 p_i [kPa]	4 Q_i	5 s_i [mixed]	6 t_i [C]	7 v_i [m^3/kg]	8 x_i [kg/kg]
[1]	-210.832	51.055	88.949	0.000	-0.074	5.000	0.001	0.424376
[2]	-210.832	51.055	1166.917	-0.001	-0.078	4.750	0.001	0.424376
[3]	106.210	51.055	1166.917	-0.001	0.937	75.964	0.001	0.424376
[4]	405.400	37.883	1166.917	0.000	1.630	128.136	0.001	0.224376
[5]	-21.876	37.883	1166.917	-0.001	0.411	29.691	0.001	0.224376
[6]	-21.876	37.883	88.949	-0.001	0.415	29.928	0.001	0.224376
[7]	1465.841	13.743	1166.917	1.000	4.806	83.724	0.139	0.975665
[8]	141.739	0.572	1166.917	-0.000	1.038	83.725	0.001	0.424376
[9]	1313.032	13.172	1166.917	1.000	4.349	37.123	0.115	0.999600
[10]	141.503	13.172	1166.917	0.000	0.500	30.000	0.002	0.999600
[11]	141.503	13.172	88.949	0.220	0.653	-35.887	0.280	0.999600
[12]	1273.273	13.172	88.949	1.000	5.394	-12.000	1.418	0.999600
[13]								
[14]								
[15]	-69.887	55.890	1969.602	1.000	-0.790	-20.000		
[16]	-51.995		3045.767		-0.789	9.672		
[17]	-318.721		3045.767	0.000	-1.781	-5.000		
[18]	-318.721		1969.602		-1.773	-20.000		

Appendix D3: CO_2 – H_2O-NH_3 Cycle Data

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THIS PROGRAM CALCULATES THE PERFORMANCE OF A SINGLE-STAGE
ABSORPTION SYSTEM WITH CONDENSATE PRECOOLING

"The following procedures are used to convert units"

PROCEDURE TPQ(TC,P,Q:x,h,v,s)
"given T (temperature), P (pressure) and Q (quality) in SI units,
 this procedures returns x, h, v and s in SI units"
TK=TC+273.15
Pbar=P*.01
CALL NH3H2O(128,TK,Pbar,Q:TK,Pbar,x,h,s,u,v,Q)
END
PROCEDURE TPX(TC,P,x:Q,h,v,s)
"given T (temperature), P (pressure) and x (ammonia wt fraction) in SI
units, this procedure returns Q, h, v and s in SI units"
TK=TC+273.15
Pbar=P*.01
CALL NH3H2O(123,TK,Pbar,x:TK,Pbar,x,h,s,u,v,Q)
END
PROCEDURE PXQ(P,x,Q:TC,h,v,s)
"given P (pressure in bar), x (ammonia wt fraction) and Q, this procedure
 returns T, h, v and s in SI units"
Pbar=P*.01
CALL NH3H2O(238,Pbar,x,Q:TK,Pbar,x,h,s,u,v,Q)
TC=TK-273.15
END
PROCEDURE PHX(P,h,x:TC,Q,v,s)
"given P (pressure in bar), enthalpy (J/g), and x (ammonia wt
fraction), this procedure returns T, Q, v and s in SI units"
Pbar=P*.01
CALL NH3H2O(234,Pbar,x,h:TK,Pbar,x,h,s,u,v,Q)
TC=TK-273.15
END
PROCEDURE TXQ(TC,x,Q:P,h,v,s)
"given T (temperature in C), Q (quality) and x (ammonia wt fraction) in SI
units, this procedure returns P, h, v  and s in SI units"
TK=TC+273.15
CALL NH3H2O(138,TK,x,Q:TK,Pbar,x,h,s,u,v,Q)
P=Pbar*100
END
PROCEDURE PXS(P,x,s:TC,h,v,Q)
"given P (pressure in bar), x (ammonia wt fraction) and entropy s, this
procedure
 returns T, h, v and Q in SI units"
Pbar=P*.01
CALL NH3H2O(235,Pbar,x,s:TK,Pbar,x,h,s,u,v,Q)
TC=TK-273.15
END

"This procedure calculates the amount of heat that is exchanged
```

```
in the solution heat exchanger"
PROCEDURE SHX(eshx,mls,mrs,ha,hb,hc,hd:hcc,hdd,qshx)
qls=mls*(hd-ha) "amount of heat on 'left side', rich solution"
qrs=mrs*(hb-hc)  "amount of heat on 'right side', poor solution"
qmin=MIN(qls,qrs)
qshx=eshx*qmin  "eshx is the sol. hx. effectiveness"
 hcc=hb-eshx*qmin/mrs  "calculation of outlet enthalpies"
 hdd=ha+eshx*qmin/mls
END

"INPUT PARAMETERS"
"efficiencies"
eshx=0.8 "solution heat exchanger efectiveness"
f=(x[9]-x[4])/(x[3]-x[4]) "solution circulation ratio"
"temperatures"
t[12]=-12 " for state point, see diagram window"
t[10]=30
t[1]=5
tglide=T[12]-T[11]
"mass fractions"
x[9]=0.9996
x[1]-x[4]=Dx
Dx=0.20
"heat"
Qgen=30000 [kW]
"Pump Work"
Wp=0 [kW]
"Quality"
Q[12]=1
Q[10]=0.00
Q[7]=1
Q[9]=1
Q[1]=0
Q[4]=0

"governing equations"
"ABSORBER"
m[12]+m[6]=m[1]
m[12]*x[12]+m[6]*x[6]=m[1]*x[1]
m[12]*h[12]+m[6]*h[6]=m[1]*h[1]+Qabs
"DESORBER"
balm=m[8]+m[3]-(m[7]+m[4])
bala=m[8]*x[8]+m[3]*x[3]-(m[7]*x[7]+m[4]*x[4])
m[3]*h[3]+m[8]*h[8]+Qgen=m[7]*h[7]+m[4]*h[4]
CALL pxq(P[3],x[3],0:T3f,h3f,v3f,s3f)
T[7]=T3f
"RECTIFIER"
m[7]=m[9]+m[8]
m[7]*x[7]=m[9]*x[9]+m[8]*x[8]
m[7]*h[7]=m[9]*h[9]+m[8]*h[8]+Qrect
h[8]=h3f
x[8]=x[3]
"PUMP"
```

```
h[2]=h[1]

"EXPANSION VALVE"
h[5]=h[6]
"SHX"
CALL tpx(T[2],P[5],x[5]:Q5e,h5e,v5e,s5e)
CALL tpx(T[4],P[3],x[3]:Q3e,h3e,v3e,s3e)
CALL shx(eshx,m[2],m[4],h[2],h[4],h5e,h3e:h[5],h[3],Qshx)
"CONDENSER"
Qcond=m[9]*(h[9]-h[10])
"EVAPORATOR"
Qevap=m[12]*(h[12]-h[11])
"EXPANSION VALVE"
h[10]=h[11]
"OVERALL"
COP=Qin/(Wp+Win+Qgen)
checkQ=Qgen+Qevap+Wp-(Qrect+Qcond+Qabs)
pratio=phigh/plow
"SET PRESSURES"
p[1]=plow
p[2]=phigh
P[3]=phigh
p[4]=phigh
P[5]=phigh
P[6]=plow
p[7]=phigh
p[8]=phigh
p[9]=phigh
p[10]=phigh
p[11]=plow
p[12]=plow
"TRIVIAL MASS BALANCES"
m[1]=m[2]
m[2]=m[3]
m[4]=m[5]
m[5]=m[6]
m[9]=m[10]
m[10]=m[11]
m[11]=m[12]
"TRIVIAL NH3 BALANCES"
x[1]=x[2]
x[2]=x[3]
x[4]=x[5]
x[5]=x[6]
x[9]=x[10]
x[10]=x[11]
x[11]=x[12]
"STATE POINTS"
CALL txq(t[1],x[1],Q[1]:p[1],h[1],v[1],s[1])
CALL phx(p[2],h[2],x[2]:T[2],Q[2],v[2],s[2])
CALL phx(P[3],h[3],x[3]:T[3],Q[3],v[3],s[3])
CALL pxq(p[4],x[4],Q[4]:T[4],h[4],v[4],s[4])
```

```
CALL phx(P[5],h[5],x[5]:T[5],Q[5],v[5],s[5])
CALL phx(P[6],h[6],x[6]:T[6],Q[6],v[6],s[6])
CALL tpq(T[7],p[7],Q[7]:x[7],h[7],v[7],s[7])
CALL phx(p[8],h[8],x[8]:T[8],Q[8],v[8],s[8])
CALL pxq(p[9],x[9],Q[9]:T[9],h[9],v[9],s[9])
CALL txq(t[10],x[10],Q[10]:p[10],h[10],v[10],s[10])
CALL phx(p[11],h[11],x[11]:T[11],Q[11],v[11],s[11])
CALL txq(t[12],x[12],q[12]:p[12],h[12],v[12],s[12])

"CO2 Cycle"
"Refrigerant"
R$='CarbonDioxide'
"State 18"
T[18]=-20[C]
h[18]=h[17]
P[18]=pressure(R$,h=h[18],T=T[18])
s[18]=entropy(R$,h=h[18],T=T[18])

"State 15"
P[15]=P[18]
Q[15]=1
h[15]=enthalpy(R$,P=P[15],x=Q[15])
s[15]=entropy(R$,P=P[15],x=Q[15])
T[15]=temperature(R$,P=P[15],h=h[15])

"State 16s"
s2s=s[15]
P[16]=P[17]
h16s=enthalpy(R$,s=s16s,P=P[16])

"State 16"
h[16]=(h16s-h[15])/eta+h[15]
T[16]=temperature(R$,P=P[16],h=h[16])
s[16]=entropy(R$,P=P[16],h=h[16])

"State 17"
T[17]=-5[C]
Q[17]=0
P[17]=pressure(R$,x=Q[17],T=T[17])
h[17]=enthalpy(R$,x=Q[17],T=T[17])
s[17]=entropy(R$,P=P[17],h=h[17])

"Equations"
Win=1000
eta=0.85 "Efficiency of compressors"
m[15]=Win/(h[16]-h[15])
(h[12]-h[11])*m[12]=(h[16]-h[17])*m[15]
Qin=(h[15]-h[18])*m[15]
```